PROCEEDINGS
THE INSTITUTION OF CIVIL ENGINEERS

PART I
MARCH 1956

ORDINARY MEETING

15 November, 1955

Mr H. J. F. GOURLEY, Vice-President, in the Chair

The Council reported that they had recently transferred to the class of

Members

Allnutt, Alan Geoffrey, B.Sc.(Eng.) (*London*).
Crowther, Harry, M.Eng. (*Liverpool*).
Cruickshank, Douglas Dickson, B.Sc. (*Edinburgh*).
De Marassé-Enouf, Marie Léon Jean.
Farquharson, George Arthur Raymond, B.A. (*Oxon.*).
Taylor, George Alexander, M.A., Ph.D., B.Sc. (*Aberdeen*).
Wilkinson, George William.
Winchester, Lamont McIntosh.

and had admitted as

Graduates

Abbott, Ivan Michael, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Amys, James William, Stud.I.C.E.
Ardagh, David Guerin, B.Sc. (*Birmingham*), Stud.I.C.E.
Arthur, Jonathan Robin Francis, B.A. (*Cantab.*).
Banerji, Biswanath, B.E. (*Calcutta*), Stud.I.C.E.
Batty, Robert Alfred, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Benjamin, Joseph Reginald, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Bestwick, Robert William, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Birkett, George William, B.Sc.(Eng.) (*London*)
Bishop, Graham Harrisson Guy, B.A. (*Cantab.*).
Bolitho, Richard Thomas Alverne, B.A. (*Cantab.*), Stud.I.C.E.
Boyes, Reginald George Hector, Stud.I.C.E.
Bradley, Dennis, Stud.I.C.E.
Brand, Ronald Ernest, Stud.I.C.E.
Briggs, Robert, B.Sc. (*Leeds*).
Broadhurst, Frederick Peover, B.Sc. (*Leeds*), Stud.I.C.E.
Brown, Frederick John.
Bulmer, Thomas Dennis, Stud.I.C.E.
Coleman, Derrick Alfred, Stud.I.C.E.
Collett, Donald Barry, B.E. (*Western Australia*).
Corlett, Derek Henry, Stud.I.C.E.
Cormack, John Mitchell, B.Sc. (*Glasgow*).
Cran, Edward Gray, B.Sc. (*Aberdeen*), Stud.I.C.E.
d'Arcy, Raymond Stanley, B.Sc. (*Wales*).
Dempster, Robert Burns, B.Sc. (*Glasgow*), Stud.I.C.E.
Edwards, Hugh Albert Morgan, B.Sc. (*Wales*), Stud.I.C.E.

Faulds, John McMillan, Stud.I.C.E.
Finch, Michael Thomas Newton, B.Sc. (Eng.) (*London*).
Fleming, William John, Stud.I.C.E.
Franklin, John Sydney, Stud.I.C.E.
Gogate, Madhukar, B.E. (*Poona*).
Green, Jack, B.Eng. (*Liverpool*), Stud. I.C.E.
Hamblen-Thomas, Edwin Oliver, B.A. (*Cantab.*), Stud.I.C.E.
Heywood, David, B.Sc. Tech.(*Manchester*), Stud.I.C.E.
Ingham, Norman John Rigg, Stud.I.C.E.
Inwang, Isaiah Isaac, B.Sc. (*Wales*), Stud.I.C.E.
Jones, Arthur William, B.Sc. (*Wales*), Stud.I.C.E.
Jones, Michael Champion, B.Sc. (*Bristol*), Stud.I.C.E.
Kenyon, William John, B.Sc.Tech. (*Manchester*), Stud.I.C.E.
Kirch, James Burton, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Knight, Derek, John, B.A. (*Cantab.*), Stud.I.C.E.
Kwaku, Samuel Francis, B.Sc. (*Bristol*), Stud.I.C.E.
Larkin, Richard Neville, B.Sc. (*Glasgow*), Stud.I.C.E.
Lee, Fulton, B.Sc.Tech. (*Manchester*), Stud.I.C.E.
McGowan, Robert, Stud.I.C.E.
McIlwaine, David, B.Sc. (*Belfast*), Stud. I.C.E.
Main, James Mein, B.Sc. (*Aberdeen*), Stud. I.C.E.
Maslowicz, Wladyslaw, B.Sc.(Eng.) (*London*).
Masterman, Ivor John, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Meakin, John Christopher.
Miller, Eric, B.Sc. (*Durham*), Stud. I.C.E.
Milsom, John, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Munro, James.
Murphy, Allan Michael, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Owens, Robert James, B.Sc. (*Durham*).
Parkinson, David John.
Pawulski, Jerzy Kazimierz, Stud.I.C.E.
Polding, Hubert Peter William, B.Eng. (*Liverpool*), Stud.I.C.E.
Pringle, John MacKintosh, Stud.I.C.E.
Raper, Brian, B.Sc.(Eng.) (*London*), Stud. I.C.E.
Redlich, Charles Patrick Basil, B.A. (*Cantab.*).
Robinson, Leonard Green, Stud.I.C.E.
Samuel, Peter Gordon, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Sellers, James Alexander, B.Sc.(Eng.) (*Natal*).
Shaw, Colin Arthur, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Shaw, Frank Roe, B.Sc. (*Nottingham*), Stud.I.C.E.
Sheppard, William John, B.Sc. (*Belfast*), Stud.I.C.E.
Sim, James, B.Sc. (*Glasgow*), Stud.I.C.E.
Stuart-Jones, Neville Francis, B.Sc. (Eng.) (*London*).
Taylor, George Mowat Brown, Stud. I.C.E.
Tippen, John.
Turner, Rupert Graham, B.Sc.Tech. (*Manchester*).
Watts, Anthony Edward, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Westwood, Peter Donald, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Whatmoor, Richard Herbert, B.Sc. (Eng.) (*London*).
Willson, Barry Ian, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Wimbush, Antony Michael, B.A. (*Cantab.*).
Wong Cheuk Chi, B.Sc. (*Hong Kong*), Stud.I.C.E.
Wong Mang-Ki, Stud.I.C.E.
Wood, John Mounsey, Stud.I.C.E.
Woods Ballard, Timothy John, B.A. (*Cantab.*).
Wright, David Ellis, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Young, Brian Sydney.

and had admitted as

Students

Andrew, John.
Barrett, Alan Dennis.
Berry, Jihl Enderby.
Blackett, Eric.
Bowler, David Allan.
Bridgman, Timothy George.
Brown, Selwyn Stanley.
Chandrasekera, Lakshman de Fonseka Sri.
Cooper, Anthony Thomas.
Copus, Graham David.
Cornish, Michael Alan.
Dadina, Kersi Nanabhoy.
Davies, George Thomas Mitchell.
Deegan, Michael.
Donohue, George.
Downey, Stanley Edward.
Duncan, Alastair.

Edmonds, Derek James.
Eze, Aja.
Fawke, John David.
Findlay, David James.
Gardiner, Barry John.
Gill, Roy.
Goddard, Clive Harold.
Gosling, Colin John.
Guiver, Victor James.
Hall, Richard John.
Hampson, Roger William.
Hancock, Peter Thomas.
Harrison, Peter.
Hellewell, David Scott.
Hifatt, John Roderick.
Holden, John William.
Howe, Michael Dean.
Hughes, Antony Brian.
Hughes, Ian Duncan Wilkinson.
Hutchinson, Bruce George.
Ireson, John David.
Jones, Gwilym Rhys.
Jones, Morris William.
Keal, Frank Edward William.
King, Christopher David.
Kinkead, Robert Andrews Hamilton.
Kouliguian, Tateos.
Laycock, Michael.
Lechem, Brian Laurence Gordon.
McLachlan, Alexander McDonald.
Mahalingam, Valautham.
Maini, Bishen Dev.
Mayfield, Denis.
Moorhouse, Brian.
Munds, Allan John.
Murphy, Francis George.
Mylchreest, Haydn John.
Newell, Donald.
Orton, Brian.
Overton, Peter George.
Park, James William.
Peters, William Gerald.
Rigby, Albert.
Robinson, John.
Russell, Anthony William.
Salmons, David George.
Sanderson, John Reed.
Taylor, Thomas Spencer.
Turner, Kenneth.
Weir, Hubert Godfrey Macoun.

The Chairman then adjourned the meeting, announcing that the remainder of the proceedings would take the form of a Joint Meeting of the Institutions of Civil and Mechanical Engineers. He invited Mr W. B. Shannon, Chairman of the Steam Group Committee of the Institution of Mechanical Engineers to take the Chair.

JOINT MEETING WITH

THE INSTITUTION OF MECHANICAL ENGINEERS

15 November, 1955

WILLIAM BOYD SHANNON, M.I.C.E., M.I.Mech.E., in the Chair

The following Paper was presented for discussion and, on the motion of the Chairman, the thanks of both Institutions were accorded to the Authors.

Paper No. 6114

THE DEVELOPMENT OF A MECHANICAL-DRAUGHT WATER-COOLING TOWER

by

*** Lawrence Gilling Smith, M.I.C.E., and Gerald Johnstone Williamson, B.A., M.I.Mech.E.**

SYNOPSIS

The Paper describes the development of a mechanical-draught water-cooling tower containing a number of novel features. The packing is composed of serrated timber laths down which the water flow is film-wise to avoid splash formation. An earlier Paper [1] gave the results of small-scale tests which indicated the theoretical advantage of grid film-flow packings in that the ratio of heat transfer to pressure drop is greater in this type of packing than in any other. A non-splash multi-trough water distributor is used to avoid the carry-over of splash droplets and hence to prevent a local deposition nuisance. Special attention has been given to the even distribution of water and air. Induced draught effects this better than forced draught and, since the fans exhaust heated air, there is no danger of ice formation upon them in frosty weather. The fans are large slow-running and aerofoil bladed, designed to give the highest possible static efficiency and to reduce the loss of velocity head at exit. The main design variables of air velocity and air/water ratio are determined from a balance of capital charges, packing, and power cost.

Several examples, showing the methods of construction, are given of these towers. The main structure and shell is usually in reinforced concrete because of its rigidity in supporting the fans and drives and its permanence and fire resistance, but timber is quite satisfactory and cheaper. The pond is in concrete in all large towers. Timber is the usual material for the packing and distribution troughs, and the water flow is controlled by orifice tubes now made in glass. Methods have been developed for supporting the laths and troughs so that they remain level and in proper relation to each other after wetting. The packing is usually built on reinforced concrete beams across the main structure.

Running experience obtained during 6 years is given together with the results of a

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Mr Williamson is Section Manager, Chemical Engineering Research Section, Imperial Chemical Industries Ltd, Billingham Division, Stockton-on-Tees.

[1] The references are given on p. 103.

comprehensive programme of tests. The good small-scale packing performance has been confirmed on the large-scale. The tests have shown deficiencies in gear-driven fan installations, established the harmful effect on cooling efficiency of recirculation of the warm exit air and, in general, put on record performance details for critical assessment of this new development.

REASONS FOR UNDERTAKING DEVELOPMENT WORK

The Authors have been associated for a number of years with the provision of cooling-water services in chemical plants. The Imperial Chemical Industries' factory site at Billingham was developed in the late 1920s, and its proximity to the tidal reach of the River Tees (mean annual fresh-water flow about 8,000,000 gal/hour) favoured the choice of river cooling water, particularly because earlier small-scale circulating cooling plant, consisting of spray ponds and forced-draught cooling towers, had resulted in high summer water temperatures with the former and a heavy fan power consumption with the latter. Two pumping stations, together capable of supplying a maximum of 10,000,000 gal/hour (averaging 50% sea water), provided most of the cooling-water requirements for more than 25 years. In addition, two concrete-shell natural-draught towers built in the 1930s gave satisfactory service, apart from a carry-over nuisance.

During the war-emergency constructional period the cooling water in the various "shadow" factories was supplied by mechanical-draught cooling towers because of the requirements of speed of construction, low initial cost, economy of timber, and camouflage; they were of the forced-draught splash type universal at that time with, in certain cases, concrete instead of timber shells. It was found that many of these towers—especially the large installations—failed to perform their specified cooling duty. Further investigations showed that the forced-draught arrangement of blowing air in at the base of the tower produced uneven air distribution through the packing and also encouraged recirculation of the warm exit air back into the down-wind fans. The fans and their housings were crude in design, and there was often a lack of air owing to an underestimation of the resistance of the packing; there was a tendency for ice to build up on the fan blades in frosty weather, and a carry-over nuisance existed in the neighbourhood of each tower.

Manufacturers' design methods were clearly rule-of-thumb. Water cooling was not treated basically as a diffusional process involving the simultaneous transfer of sensible and latent heat, and no attempt was made to determine the air velocity through the tower or the correct air/water ratio from the relation of power costs to capital charges. It was also of interest that whereas in the design of absorption towers and gas washers in the chemical industry film-flow packings were widely used, the cooling-tower industry had adhered almost without exception to the use of splash packings. On paper it was clear that the film-flow grid packing had attractive possibilities for mechanical-draught water-cooling towers; they had been demonstrated in practice by Hutchison and Spivey[2] in a number of actual installations and useful comments were also made by Cave-Browne-Cave.[3]

The large new requirements of Imperial Chemical Industries in cooling water for the post-war constructional programme at Billingham clearly could not be supplied satisfactorily from the Tees tidal water. Moreover, the river water temperatures had increased during the years because more factories were making use of it. The river water, however, caused heavy fouling and corrosion of heat exchangers; this became the over-riding factor in favour of cooling towers, because of the lower capital and maintenance costs of heat exchangers working on cleaner (cooling-tower) water.

Natural-draught towers were unattractive because of their high capital cost and because chemical-plant economics demanded a close approach to the atmospheric wet-bulb temperature. It was decided to concentrate on mechanical-draught towers and to undertake certain development work to hasten and supplement the manufacturers' efforts to improve them.

INVESTIGATIONS PRIOR TO FULL-SCALE DESIGN

Preliminary investigations started soon after the 1939–45 war on various aspects of the problem and, although some were incomplete before orders were placed for the first full-scale towers in 1948, it is convenient to consider them here before dealing with full-scale experience.

The packing (tower fill)

The use of film-flow packings in chemical plant for absorption towers indicated that they had possible advantages over splash packings in giving a higher rate of heat transfer for a given expenditure in air-pressure drop. Offset against this was the bigger usage of timber because there was no secondary surface in the form of splash. The final choice was a criss-cross (grid) arrangement of timber laths 2 in. deep × $\frac{3}{8}$ in. thick, with individual laths spaced at 2-in. pitch. In order to maintain complete film-flow the underside of the lath was serrated. In collaboration with certain cooling-tower manufacturers suitable methods of assembly were developed and tested; the emphasis was on cheap mass-production of the component parts.

The heat-transfer/pressure-drop characteristics of the packing were measured first in a small 18-in.-sq. water-cooling tower and later in a 4-ft-sq. tower. Some of this work has been described elsewhere,[3, 4] but the final data on the 4-ft tower have been slightly revised since the earlier publications and are given as lines in Fig. 5, where they form the basis against which the data from the full-scale towers are compared.

Economic design

A very useful design procedure was the establishment of easy methods of calculating the leading design dimensions of a tower in terms of the duty required, the packing characteristics, and the unit costs of power and capital. One form of this procedure is given in Appendix F of reference 4 but, for the sake of completeness, a method based on more accurate performance data and up-to-date costs is given briefly in the Appendix to this Paper. For conditions on a continuously operating plant with a power cost of 0·85*d*/kWh the economic air velocity (referred to the empty tower) is 7 ft/sec for the 2-in. grid packing, and the economic air/water ratio is such that a tower for all normal duties will require a packed height in the range 9 to 13 ft. The higher values apply to high shell costs, e.g., for an elevated tower, or to low marginal power costs, e.g., when the water pumping head is fixed independently of tower height. Such methods greatly facilitate the design of towers and enable a customer to evaluate many schemes without reference back to the manufacturer for new tenders. Moderate departures from these economic values, say by ± 20%, do not appreciably affect the final cost of the cooling water.

Water distribution

A non-splash distributor was developed of such a type that, in conjunction with ne non-splash grid packing, the carry-over of water droplets was small enough to make an eliminator unnecessary. The fact that the precipitation nuisance in the eighbourhood of a cooling tower results almost entirely from the carry-over of plash droplets and not from condensation from the plume of steamy air has been ecognized only in recent years. The work on this investigation was published in 948.[5]

The final basic form of the distributor is shown in Fig. 1. The water passes to the mall gutter troughs through glass tubes which contain an orifice restriction at the op end. (The orifice, which is normally within the size range 8 to 14 mm dia., liable to choke, which makes this form of distributor unsuitable for very dirty onditions. Troughs also suffer from a lack of flexibility in dealing with wide variaons in water rate. However, no better arrangement has yet been developed and perating experience, as described later, shows the design to be practical.)

During the development stage, measurements were made in experimental towers f the amount of carry-over for a range of air and water loadings. The small gutter roughs also went through various stages of improvement and the droplet carry-over ata given in Fig. 2 were determined on the distributor shown in Fig. 1. The effect f off-level in the troughs was also studied. For the standard air velocity of 7 ft/sec nd normal water loadings up to 200 gal/hour-sq. ft the carry-over under satisfactory perating conditions is not more than 0·01% of the water circulation rate. With hoked and overflowing troughs splash causes an increase in carry-over to a point

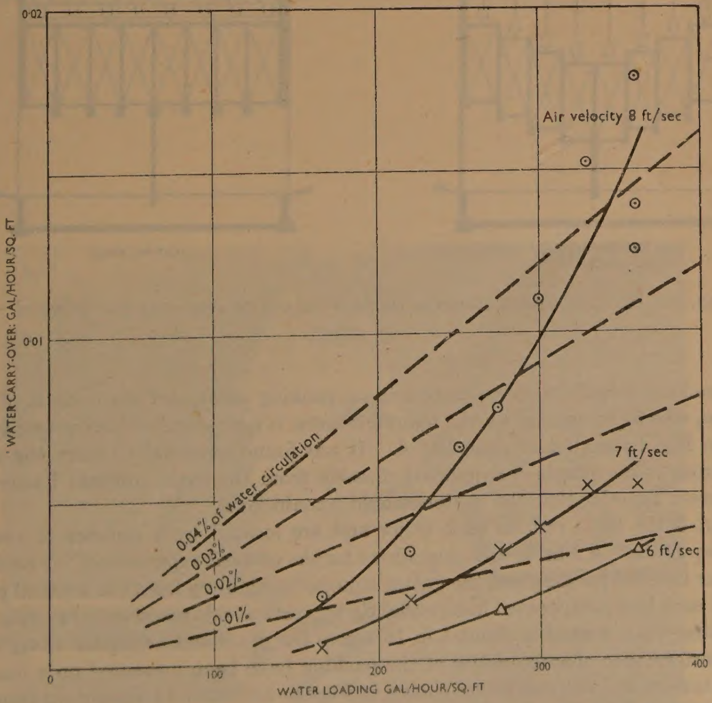


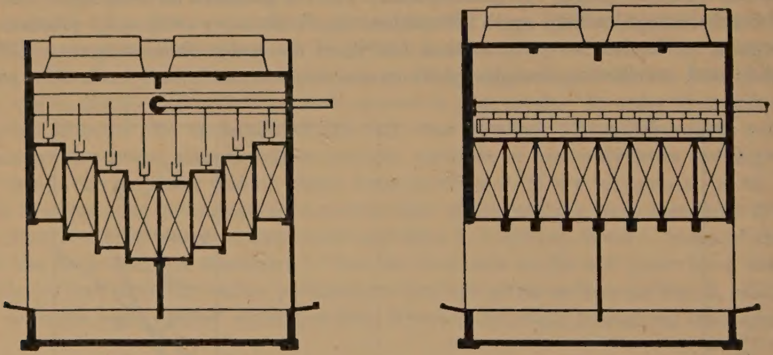
FIG. 2.—CARRY-OVER FROM NON-SPLASH WATER DISTRIBUTORS UNDER CLEAN CONDITIONS (2-IN. × 2-IN. × $\frac{3}{8}$-IN. PACKING)

where it will be very noticeable inside the tower and likely to constitute a nuisance outside the tower. (It is thought that on the full-scale towers the fans act to some extent as eliminators.)

THE FULL-SCALE TOWER

The full-scale towers of the larger type consist of one or more square cells, or units, 32 ft × 32 ft in cross-section. The air enters through openings, generally 10 ft high, in the base of each side and is drawn through the tower by fans in the roof; there are four 12-ft-dia. fans or one 24-ft-dia. fan per unit. There are similar smaller towers, of 16-ft-sq. cells with one 12-ft-dia. fan per cell. Originally a stepped packing arrangement was used in the larger units, as shown in Fig. 3a, on the grounds that it would promote more even air distribution and save headroom inside the tower. However, it was later found to have no advantage and the simpler flat arrangement in Fig. 3b was adopted.

Experience gained from operation, performance tests, and subsequent laboratory experiments is described below.



(a) STEPPED PACKING (NO LONGER USED)

(b) FLAT PACKING

FIG. 3.—PACKING ARRANGEMENTS IN GRID-PACKED WATER-COOLING TOWERS

Packing

The packing is built up in separate 4-ft-sq. packing nests, and the method of construction, developed mainly by the manufacturers, is now standardized in two types, shown in Fig. 1 (small scale) and Fig. 4. It was found essential to carry the small gutter troughs on members supported directly from the main internal framework of the tower in order that the gutters would remain level.

In Fig. 4 the laths rest on each other and are located by a number of vertical $1\frac{9}{16}$-in.-sq. posts. A 2-in. to 4-in. gap allows for the upward expansion of the packing, the upper layer of packing and the gutter troughs being hung from the vertical posts. In Fig. 1 each lath is supported independently by vertical side members (the expansion of the laths when wetted is about 1 in 16 across the grain and negligible along it).

The performance characteristics of the packing have been measured on a number of large towers and the results are shown in Figs 5a and 5b. In scaling-up from the experimental 4-ft-sq. nest to towers consisting of up to 600 of these nests, working in effect in parallel, it was thought that there would be no reduction in performance so

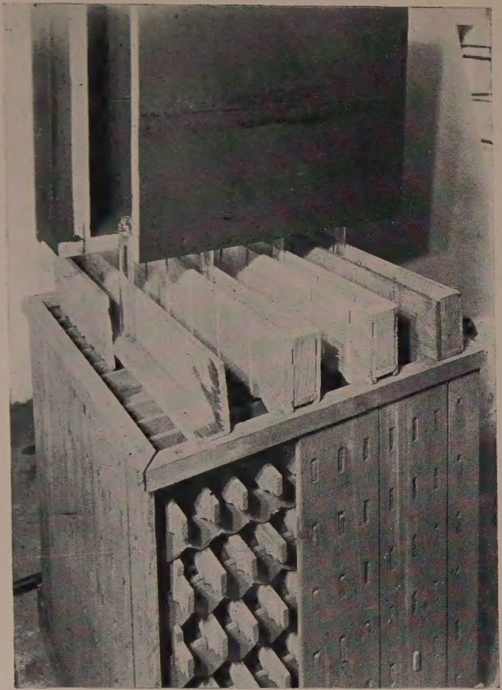


FIG. 1.—WATER-DISTRIBUTION TROUGHS

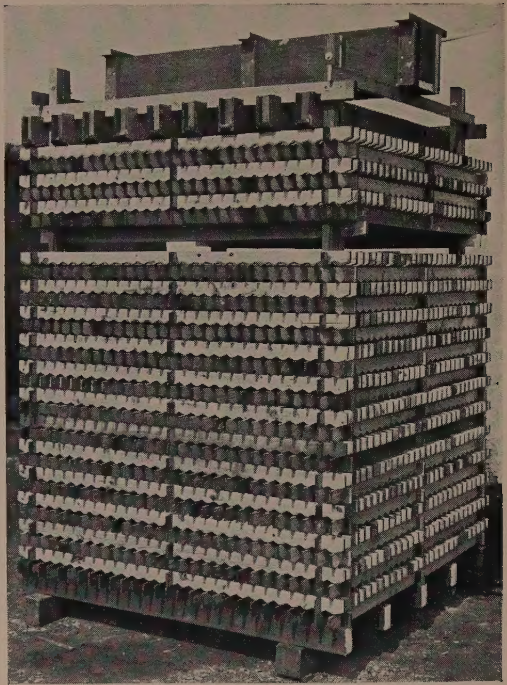


FIG. 4.—PACKING ASSEMBLY WITH INTERNAL SUPPORTS

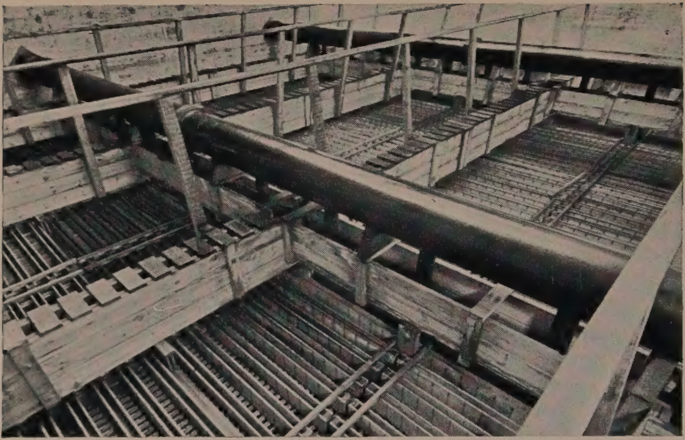


FIG. 6.—WATER DISTRIBUTORS IN A 32-FT-SQUARE CELL

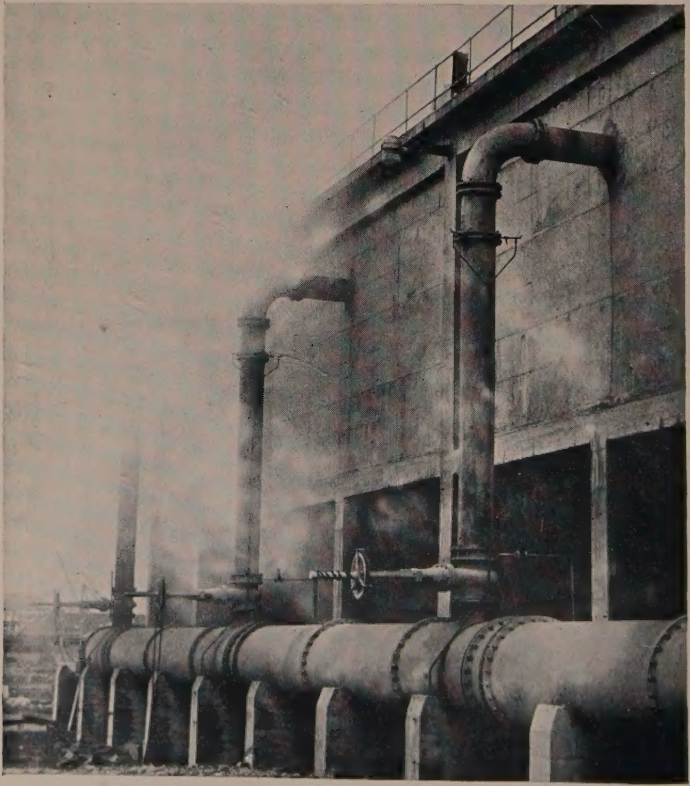


FIG. 10.—VIEW OF THE RECIRCULATION EFFECT

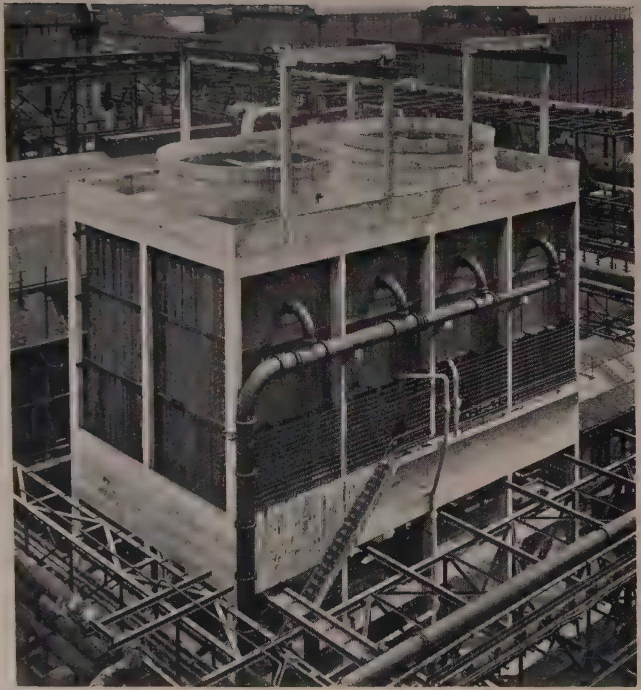


FIG. 12.—A TWO-CELL TOWER WITH TIMBER SHEETING (24-FT-DIA. FANS)

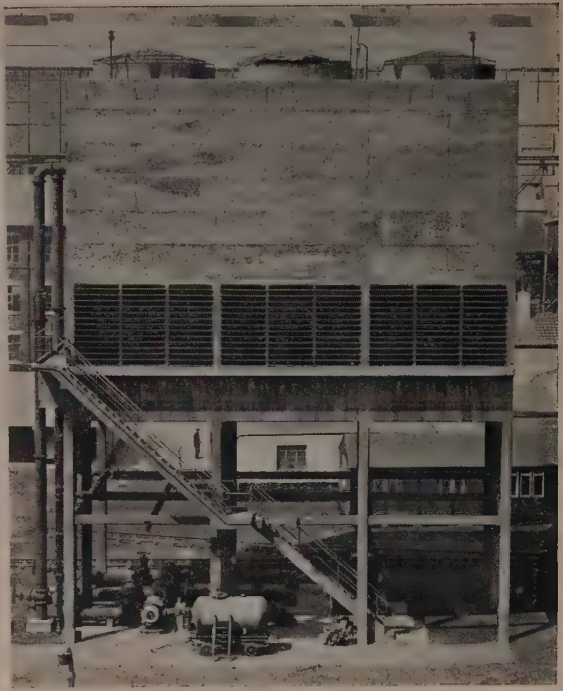


FIG. 13.—ELEVATED TOWER (12-FT-DIA. FANS)

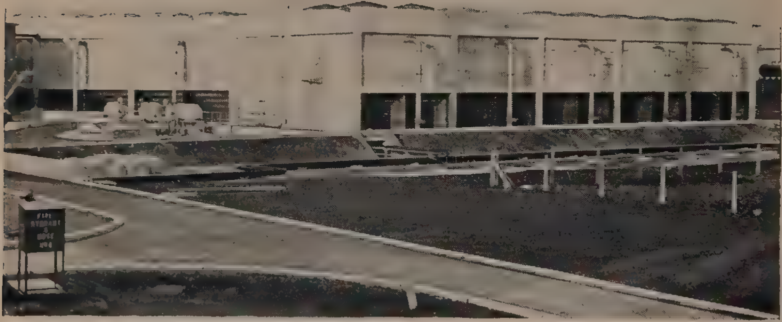


Fig. 14.—A large tower, the subject of the measurements given in Fig. 11

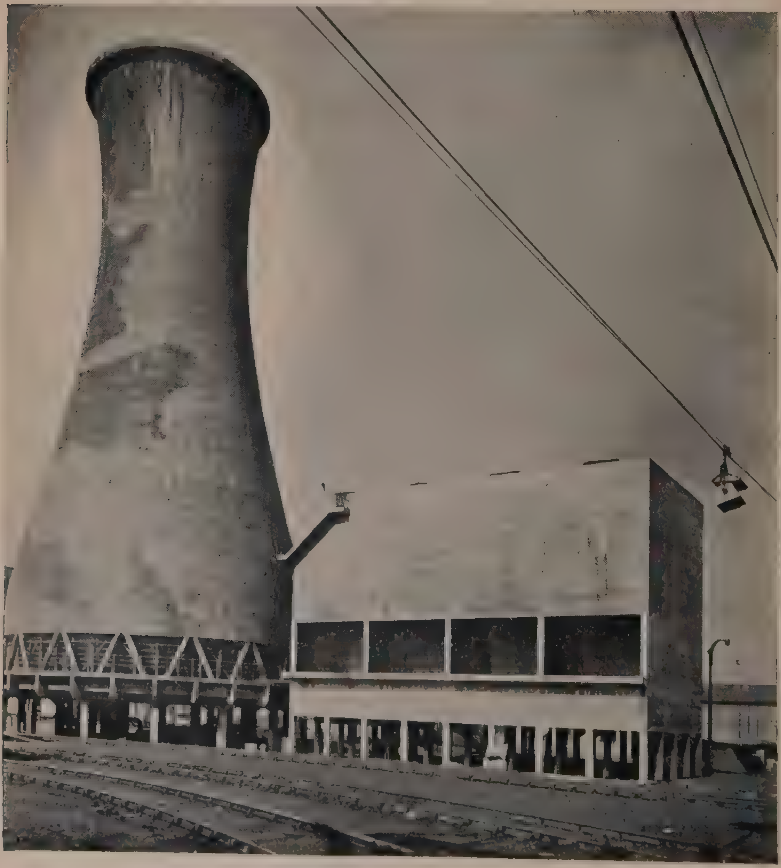


Fig. 15.—Natural- and mechanical-draught towers in series (this arrangement is suitable for a large cooling range)

long as good air and water distribution was maintained; in fact, owing to a 10-ft fall of "rain" from the packing to the pond in the full-scale tower compared with only 18 in. in the experimental tower, there should be a slight margin in hand. The transfer coefficients in Fig. 5a confirm this expectation, but show that it is necessary to maintain adequate initial water distribution. Further details of these results are given in Table 1, where data are included on the exit-water temperatures achieved compared with those specified. The methods of analysis are based on the now universally used "total-heat" method. Originally published by Merkel in 1925 [6] the method, about 15 years later, supplied a much-needed theoretical approach to the design of cooling towers, first in the United States of America and later in Great Britain.[3]

Field tests on cooling towers offer many difficulties, but during the years methods have been developed which are now quite adequate; heat balances, as shown in Table 1, are within ± 10% and often within ± 5%. After early trials with anemometers to measure air velocities inside the tower, the standard method now used is a comprehensive Pitot traverse across the top of the fan housing. The measurement of the pressure drop through the packing has presented most difficulty. In this connexion, it has always been convenient for experimental reasons to measure the pressure rise across the fan (it is usually of the order of 0·2 in. to 0·4 in. of water) and to assume that it is equal to the pressure drop through the tower, which consists of the inlet losses plus the packing-pressure drop. This in turn assumes that the static pressure on the roof of the tower is equal to that at the base. Tests were not carried out on windy days, but it is now thought that wind effects were the reason for the scatter of results in Fig. 5b. However, the design figures for the pressure drop (Fig. 5b) of five velocity heads for inlet losses and three velocity heads per foot of packing (both based on the empty tower velocity) show agreement with the tests.

Thus, the prediction of the performance of this new film-flow packing from small-scale tests has been well confirmed. Other factors of importance in determining the overall performance of a tower are dealt with below.

Water distribution

The non-splash system has been used in all towers. It has been found to work successfully even on water containing appreciable quantities of fine suspended inorganic solids, but chlorination of the water is essential to prevent biological growths, which would choke the gutter troughs. In some of the earlier arrangements the gutter troughs were not sufficiently robust nor firmly located. Breaking-away of the trough wall between notches and malalignment of the troughs caused splash, which resulted in an undesirable amount of carry-over. Improved designs of gutter trough are now used, with holes $\frac{3}{8}$ in. wide × 1 in. deep instead of notches in the trough walls; this has made the troughs sufficiently robust to be walked on. Refinements have been made to give a better initial spread of the water and to maintain the gutters in position.

The system finally adopted for the main and secondary troughs is of the type shown in Figs 3 and 6. Two main troughs run across each 32-ft-sq. unit of tower and the water enters them *via* a series of down-pipes from the two main inlet water pipes. This avoids any appreciable hydraulic gradient in the main troughs. The secondary troughs run at the same level from the main troughs; from these the water enters the gutter troughs. In order to simplify cleaning, hose connexions to a high-pressure water supply are installed and sludge plugs fitted at the end of each secondary trough.

TABLE 1.—WATER-COOLING PERFORMANCE OF 2-IN. × 2-IN. × $\frac{3}{8}$-IN. GRID PACKING

Description of tower				Performance data							Analysis of data		Remarks
Number of 32′ × 32′ units	Packed depth: ft in.	Packing arrangement	Symbol used in Figs 5 and 7		Water rate per unit cross-section $w/10$: gal/hour-sq. ft	Air rate per unit cross-section G: lb/hour-sq. ft	Water temperature: Inlet t_{w_2}: °F	Water temperature: Exit t_{w_1}: °F	Inlet air wet-bulb temperature: °F	Ratio heat to air/heat from water: %	Volume transfer coefficient $k_g s$: lb/hour-atm	Estimated re-cooled water temperature at specification water rate, cooling range, and wet-bulb temperature (specification temperature in brackets): °F	
2½	8 0	Stepped	◈	Test	131 (mean)	1,900†	Several tests			—	186 (185)‡	71·6	The first tower to be built (in 1947)
				Specification*	147	1,880	83·4	70·2	62·8	—	233 (231)	(70·2)	
1	6 0	Stepped	□	Test (1)	161	1,680	68·0	60·3	49·9	98	242 (261)	74·0	Higher re-cooled water temperatures caused by low air rates
				(2)	163	1,700	80·0	70·8	61·8	102	258 (276)	73·8	
				(3)	150	1,720	81·3	71·0	61·5	110	238 (250)	74·1	
				Specification	161	1,855	86·7	73·4	62·8	—	239	(73·4)	
3½	6 4	Stepped	+	Test	132	1,923	72·5	62·1	53·0	106	247 (242)	71·6	Test on 1½ units
				Specification	124	1,855	90·2	72·4	62·8	—	219	(72·4)	
1	7 8	Flat	▽	Test	126	1,855	75·5	57·6	41·3	94	200 (200)	74·1	
				Specification	172	1,855	89·0	73·2	62·8	—	245	(73·2)	

1	8	0	Flat	△	Test Specification	206 161	1,810 1,855	96·4 91·0	73·7 73·0	50·5 62·8	98 —	249 (252) 239	73·4 (73·0)	
4	6	8	Stepped	#	Test (1) (2) Specification	200 200 159	1,780 1,770 1,855	80·6 88·8 87·4	65·1 73·4 73·0	43·1 52·5 62·8	103 104 —	264 (272) 221 (228) 236	72·4 74·5 (73·0)	Test (1) 1952 Test (2) 1954
9	9	0	Stepped	⊙	Test (1) (2) (3) Specification	125 128 146 147	1,655 1,710 1,810 1,855	89·0 Mean of two tests 77·9 91·8	69·9 59·9 74·7	61·2 43·7 68·0	102 105 104 —	236 (258) 244 (260) 238 (240) 231	74·5 74·1 75·4 (74·7)	Tests on three individual units
1½	8	6	Flat	×	Test Specification	96 86	1,775† 1,855	133·5 103·2	73·8 69·0	56·3 62·8	— —	152 (157) 196	71·1 (69·0)	Air rate measurement impossible owing to condensation of water vapour in the Pitot tube

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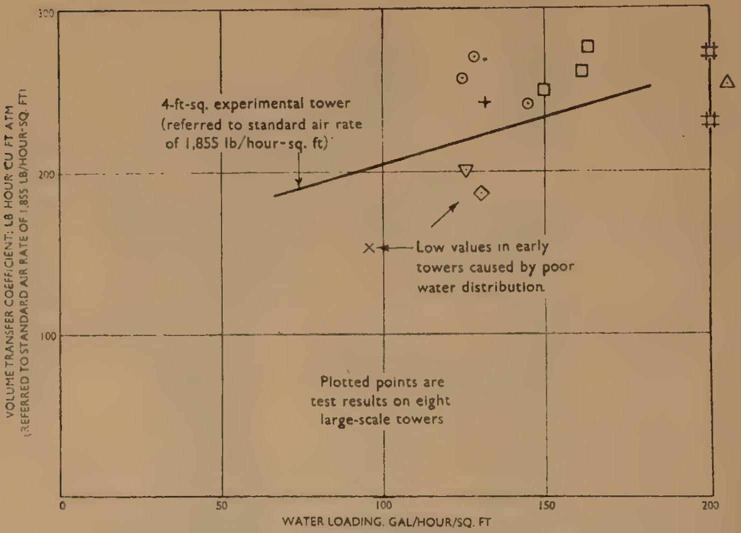
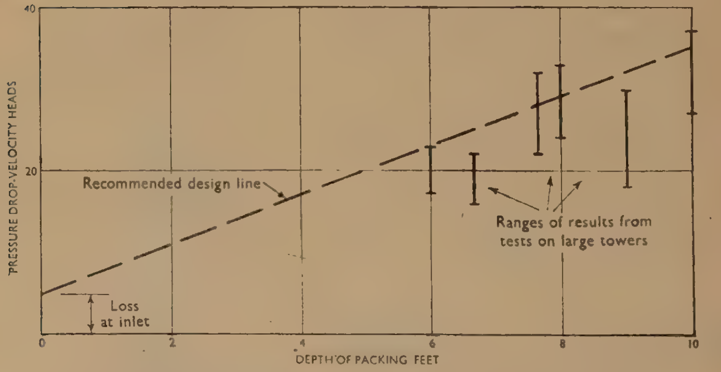
* Specification, for the purpose of this Table, means the duty predicted from the experimental tower data given in Fig. 5a. The real tower specifications included a performance margin which usually ensured adequately cooled water on the plant.

† Air rate calculated to give a heat balance.

‡ The figures in brackets are the achieved coefficients corrected to the reference air rate of 1,855 lb/hour-sq. ft. These are plotted in Fig. 5a [$k_g s\, a\, (\text{air rate})^{0.75}$].



(a) TRANSFER COEFFICIENTS FOR GRID PACKING IN SMALL AND LARGE-SCALE TOWERS



(b) AIR PRESSURE DROP THROUGH GRID PACKING IN LARGE-SCALE TOWERS

FIG. 5.—CHARACTERISTICS OF 2-IN. × 2-IN. × $\frac{3}{8}$-IN. GRID PACKING
(See also Table 1)

A subsidiary development was the use of the non-splash distribution system with shallow, 1 to 2 ft, depth of grid packing installed above an existing splash packin to prevent a carry-over nuisance. Such an installation in a power station natural draught tower has proved to be quite successful. In old forced-draught types o tower it was only partially successful because the very uneven air distribution le to excessive local air velocities with resultant stripping of the water films. Also i one particular natural-draught tower where the average water loading was as lo as 40 gal/hour-sq. ft the device, whilst successful in preventing carry-over, was quit unsuccessful as a distributor because of the difficulty in levelling the gutter trough to cope with such low rates and, also because a growth of algae high up on the walls of the concrete shell dried out and peeled off in felt-like sheets which choked the gutters. (There is no similar growth in mechanical-draught towers.)

The use of the non-splash distribution system and a shallow depth of grids merely as a splash eliminator has now been superseded by the use of eliminators above the splash-distribution system. Suitable designs of eliminator have been described by Chilton[7] and are widely used by the Central Electricity Authority; they have a very low pressure drop (about five velocity heads) and are much cheaper than the device described above. However, this device is still preferred for splash towers where the induced-draught arrangement exists; there is an addition to the heat-transfer surface which is not present with an eliminator.

Fans and drives

The induced-draught (fan-in-roof) arrangement was chosen instead of forced-draught in order to obtain an even air-flow distribution through the packing, to reduce recirculation, and to prevent ice formation on the fan blades in winter. Corresponding to the standard empty tower air velocity of 7 ft/sec the mean air velocity through the fan disk is 16 ft/sec; this is found to be a satisfactory compromise between a high air velocity to minimize fan size and recirculation and a low air velocity to minimize rejection head loss.

Experience has been limited mainly to two fans, each four-bladed and made in laminated hardwood; the 12-ft-dia. fan, in which each blade is constant in chord shape and incidence angle from root to tip and the 24-ft-dia. fan, in which the blades are twisted in aeroplane propeller fashion. Variation in duty is obtained by variation in speed. A stainless-steel fairing on the leading edge was found occasionally to be the cause of cracking in the cellulose plastic protection at its joint with the metal, leading to failure of the casein glue. It was necessary in the earlier splash towers to have metal protection against large spray droplets but the fairing is now being omitted. The manufacturers have also carried out successful tests with a synthetic water-resistant glue of the epoxy type. The fans on the whole are giving satisfactory service (the first of this type were installed in a splash tower 20 years ago) and a regular maintenance inspection is now carried out on the 150 12-ft- and twenty 24-ft-dia. fans now in service to detect and repair any minor failures in the protective coating before serious deterioration occurs.

Methods of measuring the air rate have been gradually developed to give excellent reproducibility and, according to the good overall heat balances mentioned previously, a very satisfactory absolute accuracy. Anemometers were found to be unsatisfactory in damp atmospheres and even with no water on the tower the presence of the troughs gave disturbed flow patterns which made accurate measurement difficult. Checks between Pitot and anemometer, carried out under suitable conditions during the construction of a tower, were in good agreement, the Pitot readings being 2–5% higher. Rather surprisingly the best method of air-rate measurement was found to be a four-diameter traverse by Pitot, level with the top of the fan housing. A test on a 12-ft-dia. fan requires 120 individual readings; a weighted average is taken over the very uneven flow distribution and an allowance made for the helical nature of the flow. The static pressure rise across the fan is measured by the method recommended by the Fan Manufacturers' Association.[8] The static efficiency of the fan is defined as the ratio of the power supplied to the air (air rate × static pressure rise) to the power input to the fan. In the case of fans driven directly by slow-speed motors the latter power is readily obtained. In the case of gear-driven fans the reduction-gear efficiency is unknown and is included in the static efficiency measurement. Static efficiencies are plotted in Fig. 7 and it is of interest that, for both sizes of fan, efficiencies for the gear-driven fans fall about 25% below those for the direct-driven fans. It is inferred that the transmission efficiency of the gear drives is only 75%. Experience has so far been confined to worm reduction gears with ratios in the range 4 : 1 to 9 : 1. Investigations on typical boxes for the 12-ft-dia. fan drives (6 : 1 reduction and 10 h.p.) under laboratory test conditions using a friction brake and an equivalent dead-weight load gave efficiencies in the range 88 to 90% for new and used boxes. The lower efficiencies obtained in actual operation are thought to arise from out-of-balance aerodynamic forces, which result in extra friction in the bearings; this belief is supported by the unexpectedly high maintenance on bearings. Extra friction was also confirmed by measurements of the rate of rise of temperature of the box. American experience [9] also reports some unsatisfactory service from gear

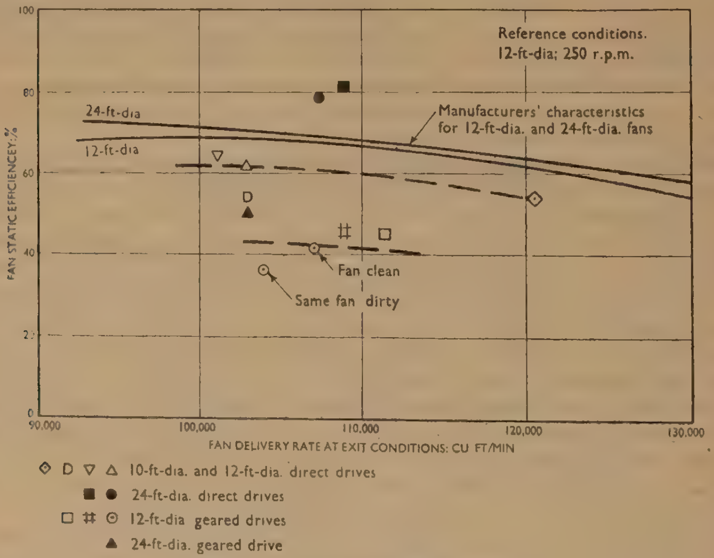
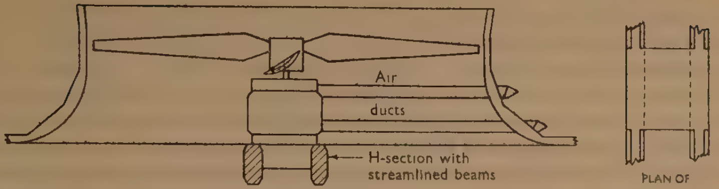


FIG. 7.—FAN STATIC EFFICIENCIES

drives on cooling-tower fans. Meanwhile the manufacturers are being consulted and until suitable drives are available it is intended to install the more costly but relatively trouble-free direct drives using large non-standard slow-speed electric motors.

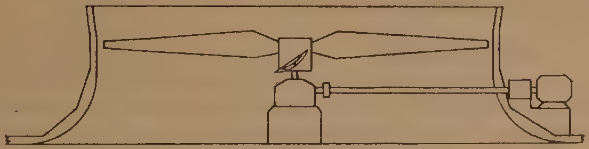
Other points of interest arising from the fan efficiency tests are:—

(1) The high efficiency of the 24-ft fans (80%) compared with the 12-ft fans (60%). The latter figure is rather lower than the specified performance and is to be expected in view of the fact that the manufacturer's specification curve was obtained on fans installed in a long smooth duct. The high efficiency for the 24-ft fans was unexpected. The fan-drive installations are shown in Fig. 8. A trial is to be made of a 12-ft fan with twisted blades and of a normal 12-ft fan in a housing with a radiused inlet similar to the 24-ft fan housing.

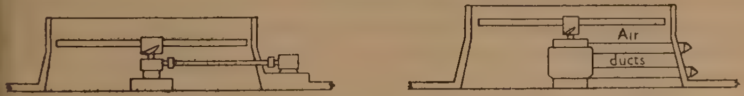


24-FT-DIA FAN WITH DIRECT DRIVE

PLAN OF
H-SECTION BEAMS



24-FT-DIA. FAN WITH GEARED DRIVE



12-FT-DIA FAN WITH GEARED DRIVE

12-FT-DIA. FAN WITH DIRECT DRIVE

FIG. 8.—FAN-DRIVE INSTALLATIONS

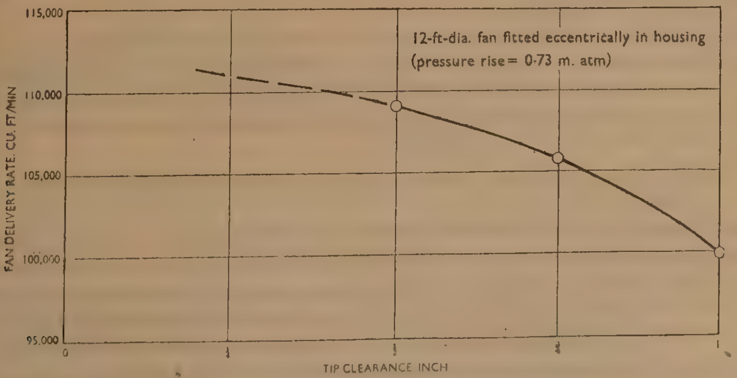


FIG. 9.—EFFECT OF FAN-TIP CLEARANCE

(2) The large decrease in efficiency which resulted when a thin layer of dirt accumulated on the upstream side of the fan blades.

Low tip clearances between the blade and the fan housing are necessary to ensure the specified efficiencies. The effect was strikingly demonstrated by air-velocity-distribution measurements on a 12-ft-dia. fan which had inadvertently been installed eccentrically in its housing. The results are shown in Fig. 9. As a good compromise between the cost of concrete shuttering and aerodynamic efficiency, tip clearances of $\frac{3}{8}$ in. ± $\frac{1}{8}$ in., and $\frac{1}{2}$ in. ± $\frac{1}{8}$ in. are specified, respectively, for the 12-ft and 24-ft-dia. fans. The supporting beams for the fan arrangements in Fig. 8 form an H in plan and are streamlined in cross-section. The cruciform plan originally adopted led in certain cases to a throbbing noise not noticeable near a tower but objectionable at a distance of more than a mile. It was attributed to the four blades passing over the four arms of the cross simultaneously.

Recirculation and the siting of towers

Recirculation of the warm air is an effect which can seriously reduce the performance of a tower. No significant amount of recirculation has been detected on the smaller towers consisting of one or two units, but in cross-winds of about 8–10 m.p.h. (the worst condition) on the larger towers the effect can be serious. Fig. 10 shows recirculation occurring and actual measurements were obtained on the same tower, which consists of a five-unit block (160 ft long) joined to a four-unit block by a high wall (the wall was necessary to complete the gas curtain formed by the towers as a fire-prevention measure), as shown in Figs 11 and 14. Perhaps because of its shape, recirculation was higher on this tower than for a straight tower; the figures shown in Fig. 11 were some of the worst measured on any tower. The effect of the various amounts of recirculation on the cooling performance is given in the form of a curve. A solid central partition wall down to pond level was originally installed on these towers to prevent excessive air-flow underneath the packing from the up-wind to the down-wind side of the tower. This prevented any flow at all and was clearly a fault so far as cooling performance was concerned. Recirculation measurements were later carried out on model towers in a wind tunnel using carbon dioxide as a tracer, with detection by an instrument of the infra-red-absorption type. The following results were obtained:—

(1) Correlation was obtained between the model and full-scale tower.

(2) Recirculation is a function of the ratio wind velocity/fan-discharge velocity. In the design of tower (assuming infinite length) with the stepped packing arrangement as in Fig. 3a recirculation reached a maximum of 8% in a cross-wind of 10 to 14 ft/sec.

(3) Extending the air-fan housings by the equivalent of 30 ft gave no improvement.

(4) Lifting the whole tower and pond on stilts 30 ft high gave no improvement.

(5) The greatest improvement from any one alteration was achieved by allowing the air to pass through to the down-wind side of the tower, underneath the packing, but not so freely that in a high wind it could pass out at the far side, carrying water with it.* This halved the recirculation rate.

All towers now incorporate a central baffle instead of the solid partition wall or, more recently, an arrangement of splash-bar packing to fill the space from the underside of the grid packing down to pond level.* The latter device in addition to acting as a wind baffle provides extra transfer surface without the need to pump the water higher.

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* Patent applications pending.

The failure of arrangements (3) and (4) to effect an improvement was unexpected and probably arises from the fact that when the air accelerates to pass either between the fan housings, or under the tower, a suction is caused which counteracts the action of the air in reducing the main suction effect. How strong these effects are was shown by Bailey and Vincent[10] in wind-tunnel experiments on the distribution of air pressure round buildings; corresponding to the increase in static pressure on the up-wind side of a building of 0·5 to 0·8 wind velocity heads, there is a suction of 0·5 to 0·8 on the down-wind side and a suction of 0·7 on the roof.

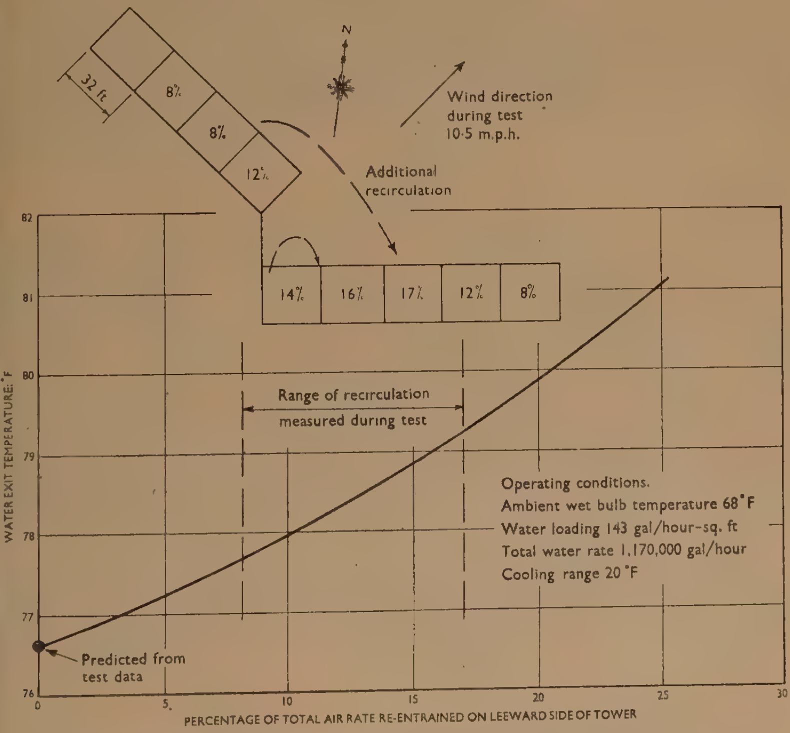

FIG. 11.—EFFECT OF RECIRCULATION ON A LARGE WATER-COOLING TOWER

In view of these downdraught effects the siting of large installations of mechanical-draught towers to avoid loss of performance from interference is a matter calling for very careful consideration. The wind-tunnel technique using carbon dioxide to determine the degree of recirculation is much to be preferred to the release of smoke or steam and is strongly recommended for this type of investigation. It is clear why, in the light of these difficulties, the designers of power-station cooling towers choose natural-draught towers; in addition to being more suitable for the easier cooling duties required in power generation they do not suffer from recirculation or emit steam plumes at low level.

The tower shell

Design has been standardized on a unit which is essentially a concrete cell 32-ft square in cross-section; the depth of packing is almost independent of the duty (it lies in the range 9 to 13 ft) and, therefore, the shell height varies little from tower to tower. The air openings are 10 ft high to give the same air velocity at entry as through the packing (allowing for its obstruction). Timber shells were replaced by concrete during the war and, in spite of the subsequent freedom to use timber, concrete has been retained except in some small towers. Where attack of the concrete might occur in places inaccessible for examination and repair, timber sheeting has been used. This type of construction is shown in Fig. 12. Completely timber shells have been known to fail in high winds but this is very exceptional and unlikely to occur if the design conforms to Code of Practice CP.112 (1952). The main objections to timber shells are unsightliness when stained with water salts, susceptibility to fan vibration troubles in the induced-draught arrangement, and fire risk when sited near a plant constituting a major fire hazard. These reasons are not well grounded in economics but, confronted with the request for an extra £1,800 per unit for concrete construction of the types shown in Figs 12–15, the management choice is usually for concrete.

Concrete shells conform to B.S. Code of Practice CP.114 (1948), and ponds and channels to the "Code of Practice for the Design and Construction of Reinforced Concrete Structures for the Storage of Liquids" (revised 1950) published by the Institution of Civil Engineers. Construction joints have a water bar. In normal conditions and good ground the pond is founded at about 2 ft 6 in. below ground level. A pond depth of 6 ft for a water depth of 5 ft gives a storage capacity amounting to 10 to 15 min supply. With this arrangement the water pumps can be placed at almost ground level and operate with a drowned suction.

Construction of the tower on stilts is a very common feature on the main factory site; seven towers of individual capacities up to 500,000 gal/hour have been built in this way. There are advantages on a congested site; the plume is discharged above the roofs of neighbouring buildings, a sheltered space is left under the pond for other plant, and the air inlet is improved. An elevated pond may permit gravity flow of the cooling water to heat exchangers, thus allowing a single pumping operation. Also, in the case of a power failure, a water flow is maintained for a short period, a very important provision in some chemical plants. Raising the tower on stilts increases the cost of the pond and shell by from 10 to 20%.

Water treatment and timber life

Little information can yet be added to what has already been published on water treatment and timber life. It is now standard procedure to chlorinate the water intermittently in order to prevent biological growths; other treatments depend on the nature of the make-up water and plant contaminants. Trials are being made with the timber packing untreated and treated with two of the modern water-soluble preservatives. "Soft rot" (chaetomium species) has been detected in one or two of the towers, and although 6 years is too short a period in which to obtain conclusive results it is clear that these preservatives have not ensured complete immunity from this type of attack. Further trials are being made with a new preservative which, it is claimed, is less liable to be leached out of the timber and is more toxic. From previous experience on about fifty cooling towers for Imperial Chemical Industries Ltd, the design has been based on a timber life for the packing (Scandinavian softwoods) of 15 years. In certain cases a shorter life than 15 years will occur. Glass laths have already been used successfully on a large scale and the use of other materials, sufficiently cheap to be competitive with timber, such as asbestos cement and thin plastic sections, is being examined.

Costs

From an analysis of the actual costs on a number of installations, unit costs have been derived which are useful for economic assessments, i.e., for comparing one scheme against another, but have to be used with caution as a basis for cost estimates. The items below show an analysis of the overall cost of a cooling-water installation; they refer to one 32-ft-sq. unit when it forms part of a large tower and include typical extras over and above the cost of the tower itself.

The figures in Table 2 emphasize the very large proportion of the total cost required for the water mains even under normal circumstances. It is this factor which makes it cheaper on a large industrial site to install smaller cooling towers at the place where the water is used than to install a large tower centrally and distribute the water to the plants concerned.

TABLE 2.—TYPICAL COSTS FOR GRID-PACKED INDUCED-DRAUGHT TOWERS

	£	£	
Design charges		2,000	
Structure, foundations, pond (all concrete) at £5·4/sq. ft	5,530		33% less for wooden structure 10–20% more for elevation on stilts
Packing (11 ft high) at 8*s* 6*d*/cu. ft	4,800		
Distributor 15*s*/sq. ft	770		
One 24-ft-dia. fan (4-bladed)	900		
Slow-speed 50-h.p. motor and starter (150 r.p.m.)	1,700		
Installation of fan and drive plus ventilation, wiring, lifting beam, etc	570		
Total for tower		14,270	
Clearing site, access, and drains	2,500		
Underground mains	8,000		
Pumping plant	5,000		All these costs can vary widely
Power distribution, lighting	2,500		
Water-treatment equipment	3,000		
Instruments	500		
Total for services, etc		21,500	
Total for installation		37,770	

Twenty-four-ft-dia. fans are used only when towers are large enough to consist of a number of units. For smaller towers, consisting of one or two units, 12-ft-dia. fans are used to give flexibility in taking fans off-line for maintenance. Direct drive is used for both the 12-ft- and 24-ft-dia. fans. An economic comparison with reduction-gear drives is given in Table 3.

In view of the lower running cost of the direct drives and the heavy, but uncertain, cost of maintenance on the gear drives, direct drive is now adopted.

TABLE 3.—COST DATA FOR FAN DRIVES

(Based on one 32-ft-sq. unit)

	Four 12-ft-dia. fans		One 24-ft-dia. fan	
	Direct	Reduction gear	Direct	Reduction gear
	£	£	£	£
Fans	700	700	900	900
Drive	—	480	—	580
Motors and starters	2,300	420	1,700	360
Erection, wiring, lifting beams, etc. . .	720	740	570	590
	3,720	2,340	3,170	2,430
Fan static efficiency: %	60	60	78	78
Motor efficiency: %	83	88	84	88
Transmission efficiency: %	100	75	100	75
Overall efficiency: %	50	40	66	52
Annual cost of power for air rate of 2×10^6 lb/hour and a pressure drop (11 ft of packing) of 0·44 in. of water (1 kWh costs 0·85*d*)	1,430	1,790	1,080	1,370

The cost calculations to determine the optimum design of tower have already been given in full elsewhere [4] but, because of changes in the cost data since 1950 and a slight difference in method, a brief recapitulation is given in the Appendix. The economic air velocity (based on the empty tower) is still 7 ft/sec but, because of the greater rate of increase in concrete costs relative to packing costs, towers tend to be thinner and taller; the economic height of packing is now about 9 to 13 ft.

No attempt is made in the Paper to carry out a cost comparison of the grid-packed tower with a mechanical-draught splash tower or with a natural-draught tower. The purpose of the Paper is to put on record data on one particular type of tower which would help to make such a comparison possible. However, calculations by the Authors using the limited data available to them show quite definitely that, for the relatively difficult cooling duties which are economic on chemical plants but not, apparently, on power-generation plant, mechanical-draught towers provide cheaper water than natural-draught towers. In the mechanical-draught field the mechanical-draught grid-packed tower with its film-flow packing was a considerable advance in design over the splash-packed forced-draught tower, especially for the larger installations, but the advantage compared with modern types of splash tower, though positive, is much less.

A general approach to the economic design of film-flow packings was published recently by Margen [11] who claimed that the method could be extended to splash packings. When suitable performance data are available on splash packings economic comparisons of the type mentioned above can then be made. The Authors consider it very desirable that such information should become available so that the customer is in a position to make these preliminary assessments; it enables him to specify, in greater detail, and to appropriate contractors, his particular requirements.

ACKNOWLEDGEMENTS

The Authors acknowledge the work of many colleagues in Imperial Chemical Industries who contributed to these investigations, particularly Mr R. C. Hutchinson, B.Sc., who assisted in the preparation of the Paper, also the work of the Davenport Engineering Co. Ltd and Film Cooling Towers (1925) Ltd for developments on packings and water distributors. (The latter is also thanked for permission to reproduce Fig. 4.)

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APPENDIX

OPTIMUM ECONOMIC DESIGN

It is assumed that the duty of the tower is given, i.e., the water rate and temperatures are fixed, but the air velocity and the air rate need to be determined. There is an infinite number of towers which will do the given duty but only one, the optimum, which results in a minimum value for the sum of the power costs and capital charges. Only those costs which influence the position of the optimum need be included in the expression for "total marginal cost of cooling water." For the purpose of the analysis these costs are called marginal; these exclude the costs of fixed items common to all schemes.

Costing procedure

The chief marginal costs depend on the tower cross-sectional area, the volume of packing, the water pumping height (or height of packing), and the air pressure drop through the tower. The data in Table 4 hold for a tower with an 11-ft height of packing and 24-ft-dia. fans, direct driven.

TABLE 4

Item	Unit cost	Annual charge			
		Return on capital: %	Maintenance: %	Total %	£
Structure, pond. Lifting beam, electrical wiring	£5·4/sq. ft £0·6/sq. ft	12½	2½	15	$6{\cdot}0 \times 0{\cdot}15 = 0{\cdot}90$
Fans, motors, starters . . . Distributor . .	£2·55/sq. ft £0·75/sq. ft	12½	7½	20	$3{\cdot}3 \times 0{\cdot}2 = 0{\cdot}66$
					$C_s = 1{\cdot}56$/sq. ft
Grid packing .	£0·425/cu. ft	12½	7½	20	$C_p = 0{\cdot}085$/cu. ft
Fan power (overall efficiency = 66%) . . .	0·85*d*/kWh	—	—	—	$C_f = 5{\cdot}0 \times 10^{-4}$/16-m.atm /hour
Water-pump power (overall efficiency=70%)	0·85*d*/kWh	—	—	—	$C_w = 1{\cdot}67 \times 10^{-5}$/lb.-ft/hour

The total marginal cost of providing w lb/hour of water is given by the sum of the following four factors:

$$\underset{\text{Total annual marginal cost}}{C_t} = \underset{\substack{\text{tower cost}\\\text{(excluding packing)}}}{C_s \,.\, a} + \underset{\text{packing cost}}{C_p \,.\, al} + \underset{\substack{\text{water pumping}\\\text{cost}}}{C_w \,.\, wl} + \underset{\text{fan power cost}}{C_f \,.\, G \,.\, \Delta p} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where a denotes cross-sectional area of the tower in sq. ft.
l „ height of packing in feet.
G „ air rate in lb/hour.
w „ water rate in lb/hour.
r = G/w lb/lb.
Δp dedotes pressure drop through the tower, m.atm.

Equation (1) can be re-arranged in the form:

$$\frac{C_t}{w} = C_s \,.\, r\left(\frac{a}{G}\right) + C_p \,.\, rl\left(\frac{a}{G}\right) + C_w \,.\, l + C_f \,.\, r\Delta p \quad . \quad . \quad . \quad (1a)$$

Because of the dependence of l on the air rate G, and the air velocity $G/\rho a$, and of Δp also on the air velocity, there are only two independent variables in this equation, namely, r and G/a. (ρ is the air density in lb/cu. ft.)

The height of packing can be calculated from a knowledge of the cooling duty required

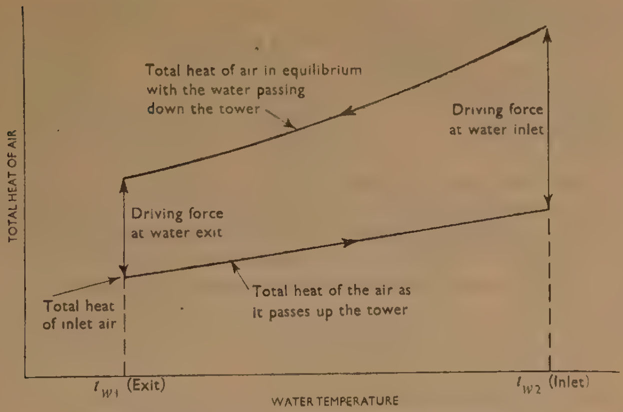
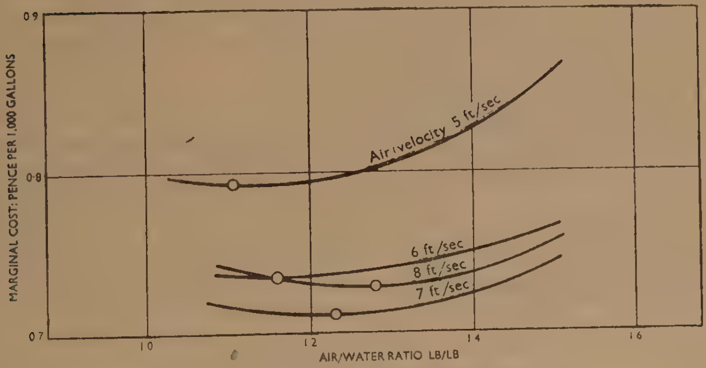
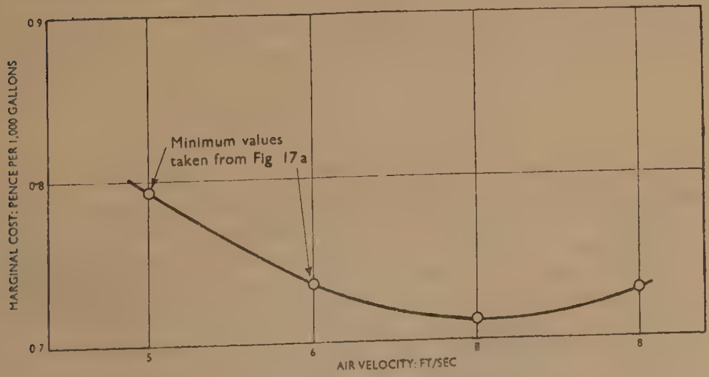

TOTAL HEAT OF AIR
Total heat of air in equilibrium with the water passing down the tower
Driving force at water inlet
Driving force at water exit
Total heat of inlet air
Total heat of the air as it passes up the tower
t_{w1} (Exit)
t_{w2} (Inlet)
WATER TEMPERATURE

Fig. 16.—Total heat transfer diagram



(a) ECONOMIC AIR/WATER RATIOS



(b) ECONOMIC AIR VELOCITY

Fig. 17

and the transfer characteristics of the packing, using the total-heat equation as developed for water-cooling analyses, namely:

$$\underset{\substack{\text{Heat lost by the}\\ \text{water and gained}\\ \text{by the air}}}{w(t_{w2}-t_{w1})} = \frac{1}{0{\cdot}623} \cdot \underset{\substack{\text{volume}\\ \text{transfer}\\ \text{coefficient}}}{k_g s} \cdot \underset{\substack{\text{volume}\\ \text{of}\\ \text{packing}}}{al} \cdot \underset{\substack{\text{mean}\\ \text{driving}\\ \text{force}}}{\Delta H_m} \quad . \quad . \quad (2)$$

w_2, t_{w1} denote inlet, exit water temperatures in °F.

$k_g s$ denotes volume-transfer coefficient (obtained from Fig. 5a), lb/hour-ft^3-atm.
ΔH_m ,, mean driving force (given by the harmonic mean of the ordinate intercepts between the two lines on a diagram of the type shown in Fig. 16), B.Th.U/lb.

The pressure drop through the tower is given by the equation:

$$\text{Pressure drop} = \text{constant (number of velocity heads lost)} \times \tfrac{1}{2}\rho\left(\frac{G}{\rho a}\right)^2 \text{ m.atm}$$

i.e.,
$$\Delta p = 0{\cdot}0147(3l + 5) \times \tfrac{1}{2}\rho\left(\frac{G}{\rho a}\right)^2 \text{ m.atm}$$

A straightforward, though rather tedious, method of obtaining the minimum value of C_t is illustrated by an example. Quicker methods are available but they tend to obscure the principles, which are all that it is desired to demonstrate in this Appendix.

Example of the determination of optimum tower design

A tower is to cool water from 87·8° F (31° C) to 69·8° F (21° C) with ambient conditions of 62·8° F (17·1° C) wet-bulb temperature.

With successive choices of air velocity of 5, 6, 7, and 8 ft/sec, the value of C_t in equation (1*a*) is calculated for various values of *r*. Fig. 17a shows the curves obtained. The slightly different shape of the curve for 8 ft/sec is due to the inclusion at velocities above 7 ft/sec of a spray eliminator—an extra five velocity heads are included in the pressure drop in equation (4).

The minimum values of C_t are then shown plotted in Fig. 17b. The optimum air velocity is 7 ft/sec and recalculation with the corresponding optimum air/water ratio of 1·23 gives a packed height of 11 ft.

Similar calculations for other cooling duties over a wide range reveal that the optimum air velocity is fairly constant at about 7 ft/sec and the optimum air/water ratios result in packing heights in the range 9 to 13 ft.

The Paper, which was received on the 6th August, 1955, is accompanied by thirteen photographs and nine sheets of diagrams, from which the half-tone page plates and the Figures in the text have been prepared, and by the following Appendix.

CORRESPONDENCE on this Paper should be forwarded to reach the Institution before the 15th March, 1956. Contributions should not exceed 1,200 words.—SEC.

Discussion

Wing-Commander T. R. Cave-Browne-Cave (Consultant) said that during the 1939–45 war he had spent a considerable time inside the cooling towers of Hams Hall and had been much impressed by the completely disorganized movement of air inside the tower. In certain places water descended with great air streams actually blowing down on top of the packing. He had been very impressed by the effect of the falling water on the air movement. He concluded that it was desirable to avoid liquid water in the upper part of the tower, and therefore to depend entirely on film contact.

After the war he had carried out some experiments at Southampton to determine whether it was possible to arrange for the water to flow vertically downwards over flat continuous sheets, covered with a film of water. He found that very high air speeds could be used over the sheets before the water formed waves and was blown off but, since experimental facilities at Southampton were not equal to those at Billingham, he had not continued his tests, but had described his recommendations and tests to Billingham.

If air at 60°F outside the tower was heated to 80°F inside the tower there was a certain decrease in density. If the air was then saturated a further decrease in density occurred. The total decrease was quite small, about 5%. If that air contained a quantity of falling water the direction of motion might be reversed, because the water suspended in the air added to the weight of the warm column, and might annul its buoyancy. In natural-draught towers that buoyancy was essential; even in forced-draught towers the buoyancy of the air when it got outside was a very useful factor to resist recirculation.

It was surprising that the most satisfactory form of film-type packing was that described in the Paper, consisting of ⅜-in. × 2-in. battens crossed and at a pitch of 2 in. Friction drag on that surface was beneficial and caused the evaporation of water and the cooling. Form drag resulting from the bad shape was of little, if any, benefit. How bad that form drag had been was shown by data on p. 91, from which he deduced that the resistance in the packing was eight times the resistance at the intake. He preferred long vertical sheets of material over which the water would flow down uniformly, and which would give useful friction drag and almost no form drag. Could the Authors say whether they regarded that as wrong or if it had been inconvenient to arrange?

With regard to propellers he thought that the difficulty experienced with the gearbox was caused by the two bearings being too close together. In the Royal Air Force at the beginning of the war efforts had been made to achieve static balance of large airscrews to 1 in.-oz, overlooking the fact that the greatest unbalanced couple on the propeller, and therefore on the machine, arose not from the out-of-balance force but from the variation of thrust between one blade and another. It arose partly from variation in shape, partly from variation in the angle at which the blade was set, and partly from inflow into the propeller. Those forces were very large.

He thought it would be wise, therefore, to carry the fan on a solid vertical shaft of considerable length, with a bearing at the bottom which would take the out-of-balance couple. The gearbox should be carried on that shaft. The efficiencies which the Authors recorded were surprisingly low; their suggestion that that might be the result of bearing trouble in the gearbox and lack of alignment was probably correct. An arrangement such as he suggested, therefore, might produce a great improvement.

It might be found, moreover, that through the centre of the airscrew a considerable amount of recirculation was taking place; that could be avoided by a boss fairing of considerable diameter. By adding a streamlined tail a closer approach would be made to airscrew practice, and would probably produce a considerable improvement.

Mr B. Wood (of Messrs Merz & McLellan) pointed out that the Authors had explained that their remarks were applicable to towers in the chemical industry and not to those for power stations. He hoped that would not be lost sight of. It was widely supposed that since the Americans used mechanical draught, the power station industry in Britain was out of date in relying largely on natural draught. In fact the position was the reverse: until recent years few large power stations in the U.S.A. had required to use cooling towers since most were sited on large rivers. Cooling towers were now becoming necessary and, since mechanical-draught towers had not proved satisfactory, there was at present a search for an alternative which might well result in the adoption of the natural-draught tower as known in Britain.

All the defects mentioned by the Authors were known in the U.S.A. Gearboxes, fans, motors, and timber gave trouble and 90% of the towers failed to meet their guarantees. Recirculation was now recognized to be serious and high chimneys had been fitted to many towers. Induced draught had not cured the recirculation trouble which was erroneously attributed to forced draught *per se*. Forced draught was accordingly too hastily condemned in favour of induced draught, which had brought with it new fan and gearbox troubles which did not occur with forced draught, as well as higher head loss, higher fan power, and higher costs. In one case known to Mr Wood the fan power in an induced-draught tower had turned out to be 50% above what it was supposed to be. The reason had not yet been fully isolated, that was, whether it lay in the fan or the gearbox. One remedy was to return to forced draught with suitable precautions to limit recirculation, for example, by letting air pass to the lee side under the tower.

He thought the Authors had fallen into the trap of thinking that what was not published was not knowledge. In dealing with carry-over they gave a reference to 1948, whereas Mr Wood had known as early as 1932/3 that the explanation of precipitation from the plume was untenable and that tests about that time had shown that the salt content of caught deposits was that of the pond water. That information was not, however, published.

Regarding economics he suggested that electricity should be charged on a two-part tariff, not on a flat rate. That could make a big difference in economic studies since, if the tower was used for even half an hour in the winter, it would incur the full demand charge of £5 or £6 per kW year. Hence load factor was generally of less account than might be supposed.

Although the Authors had said that the film-packed tower represented an advance in design, Mr Wood had not found it commercially competitive with splash-bar designs in his comparisons of tenders. The common notion that a mechanical draught tower was a great deal cheaper than a natural draught one was not borne out by tenders. Mr Wood said that when mechanical draught was offered to a specification which required recirculation to be taken into account, namely, by measuring the wet bulb away from the tower influence, then the saving would be only 15 to 20%, which was not generally enough to pay for the capitalized fan power. It would certainly not cover also an insurance against troubles with the "gadgets" which were absent with natural draught.

He thought, therefore, that for installations of say, 1,000,000 or 2,000,000 g.p.h. and upwards where it was necessary to get the vapour away at high elevation or where fan noise could not be tolerated, there was no present alternative to the natural-draught tower as used in England.

Mr H. Chilton (Research and Development Engineer, C. A. Parsons Ltd) said some of the problems which the Authors seemed almost to have taken in their stride had long been troubling cooling-tower designers. One example was the work on the recirculation of heated air, on which he would like to have more information.

On p. 98 a recirculation ratio of 8% was mentioned. Did that refer to a single row of towers with the wind at right-angles to the major axis; if so, could it be assumed that recirculation would be less troublesome if the wind was parallel to the major axis? He would be grateful if the Authors could offer some actual design figures on the assumed recirculation rates in the design of, say, single and multiple lines of mechanical-draught towers, assuming fairly good site conditions.

He thought that some circulation difficulties might have been caused by abandoning the louvered inlet in favour of a solid central dividing wall. There was much in favour of the louvered inlet, because it could be so designed to provide a net air-flow outwards through the leeward side inlet without losing any water, because the louvered inlet worked as a crude sort of eliminator. Experiments indicated that the louvered inlet might be improved by a rather lower resistance to air-flow; that would be easy to arrange by making the louvers closer to the horizontal.

In regard to the almost astoundingly low efficiency of the worm-gearboxes, it was surprising that if the load factor was high enough direct drive could be economically sound. Of course the delivery of the large special slow-speed motors was so poor that very often a contractor was prevented from offering direct drive by that alone.

The deterioration of timber packings was important in cooling-tower economics. The chaetomium species attack was a surface action, and failure occurred when the thickness of the sound timber was reduced to about $\frac{1}{8}$ in. Grid packing, with its fairly thin layers, might be supposed to fail more quickly than the more substantial timber in splash-bar packing, even allowing for the grids being much better supported. Even, however, if the same life was assumed for both types, it should be remembered that the cost of replacing the grid packing was several times above the cost of replacing splash-bar packing. Had the Authors taken that into account in their comparison?

There were, of course, other factors in favour of splash-bar packing. The timber type of shell construction, which was cheaper, tended to favour splash-bar packing, as did such factors as easy cooling duty, dirty circulating water, circumstances in which the pump head was irrelevant, and possibly low load factor. He hoped, therefore, that the Authors would agree with his suggestion that there might still be a very useful commercial field for the older type of splash packing, as well as for the new film-type packing which they had so ably developed.

Mr W. J. King-Smith (Technical Manager, The Airscrew Company & Jicwood Ltd) said that the Authors had referred to ice formation on fan blades, particularly with forced-draught cooling-towers, and had suggested that the difficulty could partly be overcome by using induced-draught fans. It seemed probable, however, that in winter cooling, when the fans would presumably be stationary for much of their working time, there might also be an icing problem with the induced-draught fan. If that was the case, it would be very interesting to know what methods I.C.I. Ltd had employed to overcome the problem. He imagined that they had probably used de-icing pastes, but information would be of interest. Had they any other method of overcoming the possible difficulty?

That led him to the question of the importance of icing on fan performance. In Fig. 7 there was a graphic illustration of the effect of dirt on fan performance. It was obvious that when the fan contour was spoilt by dirt, performance would be reduced, and ice could have the same effect. It was particularly annoying in forced-draught applications. He had seen an installation at a gasworks where there was a thick hard crusted deposit on the blade which had to be chipped off, and which had been 1/16 in. thick or more. What degree of dirtiness gave the 4 to 5% reduction in performance referred to by the Authors? He would like to see how it compared with what his company had found from experience. Again, with regard to drop in efficiency, had that been brought about by increased power input, as well as by reduction in performance, or was it from one cause only?

Mr Williamson had mentioned the difficulty of obtaining satisfactory results in static pressure measurements. Mr King-Smith was interested in the point because he had intended to ask the Authors where they had actually measured their static pressure in the towers and whether they had selected those points as a result of laboratory work done on their model tower. Clearly the actual siting of the points must have had a big influence on the results obtained.

Why had the Authors chosen four-bladed instead of two-bladed fans? There was, of course, a difference, though not very large, in efficiency. In forced-draught applications two-bladed fans had sometimes given severe trouble owing to wave formation in ponds, but the same disadvantage did not apply to the induced-draught fan. He could only assume that four blades had been used to obtain a more even distribution through the packing than occurred with the two-bladed fan.

In connexion with the difference in efficiency between the 24-ft four-bladed fan and the 12-ft four-bladed fan—not so much the absolute difference in efficiency but the way in which it fell short of the manufacturers' curves, as given in Fig. 7—he thought that difference could be accounted for solely by the difference in design of the tower outlet. From Fig. 8 the 24-ft-fan tower outlet was well flared and shaped, but the 12-ft-fan outlet was almost straight, with only a slightly chamfered inlet to the fan. That difference could account for the difference in performance.

With regard to power loss from gearboxes and whether it was contributed to by aerodynamic out-of-balance or some other cause, he did not know the answer but would be interested to hear what gearbox manufacturers had to say. In the experience of his company, which was not quite the same as that of the Authors, they had found that the trouble lay far more with the worm-wheel than with the bearings. There might be many reasons for that, but he thought much of the trouble might be due to bad lubrication from one cause or another. From the diagrams in the Paper it appeared that the motor-driven arrangement was not basically different so far as the bearings were concerned from the gearbox arrangement. Although the bearings might be a little farther apart, the difference was not sufficient to decide between success and failure. Breather pipes were shown, however, from the motor to the outside of the tower, but there were no analogous pipes from the gearbox. He thought there might be some type of emulsification of the lubricating oil which might affect performance. He would be interested in any suggestions on those lines, because he had always found that it was almost impossible to attribute the extra load to the fan. Fan design was a fairly exact science, and little miscalculation was made with powers required, but vastly lower efficiencies on the gearbox and drive assembly did occur.

Mr G. Ripley and **Mr R. C. Hutchinson** (Technical Engineers, Davenport Engineering Co. Ltd) felt that it would be interesting to compare the economics of splash-bar and film-flow packings. Fig. 18 showed the characteristics of a typical splash-bar fill. The results had been obtained from a 5-ft-square experimental tower and represented the characteristics of splash packings in general. Although several packings had been tested in the tower, none had been found to have an overall efficiency better than the others. The transfer coefficients had been confirmed in a large tower, although the pressure loss was lower in the field test than on the small experimental tower; that was due, no doubt, to the difficulty of taking measurements in the large tower, particularly since the tests had been undertaken when fairly strong winds prevailed.

Using the characteristics shown in Fig. 18 and the information in the Paper, it became possible to design both a splash-packed and a film-flow tower to undertake a specified duty. The comparative sizes of the towers were shown in Fig. 19. The basis of the comparison between the two types of packing was the assumption that the towers were to work continuously throughout their operational life. Furthermore, it had been assumed that the shells of the towers were of reinforced concrete. The splash-packed tower was an economic design in the sense that the operating air velocity was 6 ft/sec and the air/water ratio had been derived from cost considerations as applied to the film-flow tower. That accounted for the obvious increase in the cross-sectional area of the two types of packing.

Using the information in the Paper, it was possible to compare the capital and running costs of the two towers. In relation to costing, a point not mentioned by the Authors was the cost of the splash-bar packing, which Messrs Ripley and Hutchinson had assumed as 2*s*/cu. ft. Fig. 18 showed that the capital cost of the splash-bar-packed tower was slightly less than that of the film-flow tower, but when running costs were considered the film-flow type was easily the better. The comparison obviously favoured the film-flow packing because the operational cycle was continuous; there was, however, a definite case for splash-bar packings, and he would like to mention a few instances.

The first was where an industrial plant had to operate only in a short cycle, probably 8 hours a day and 5 days a week. In that case the velocity of the air through the packing could be increased, with the consequence that capital expenditure on the plant would be less. Admittedly the running costs per hour would be higher, but that would not have the same effect in the conditions envisaged as with continuous operation. Secondly, on many gas stations the water was fed by gravity from the condensers direct to the cooling tower, so that running costs of the two types of plant came closer together. Finally, where dirty water was used in cooling towers film-flow packings were not advisable. Since the distance between the laths was only 2 in., there was a possibility of bridging over with dirt and the trough distributors readily became clogged. As soon as that occurred there was a marked loss in efficiency, whilst in addition mechanical carry-over could be a nuisance. It should be remembered that eliminator screens were not usually fitted to film-flow towers.

In conclusion, they would advise all purchasers of cooling towers that it was essential to give the manufacturers the true operational cycle of the tower, and to state what type

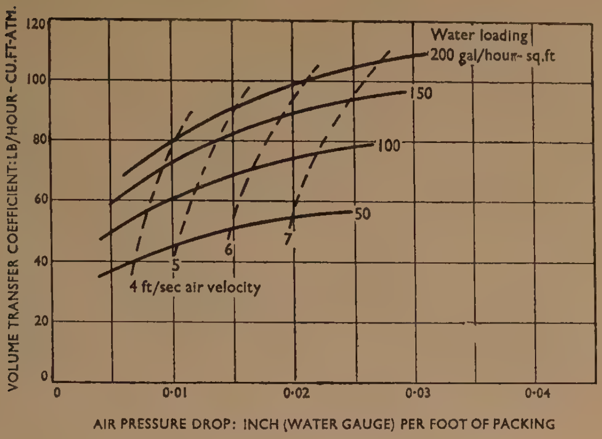
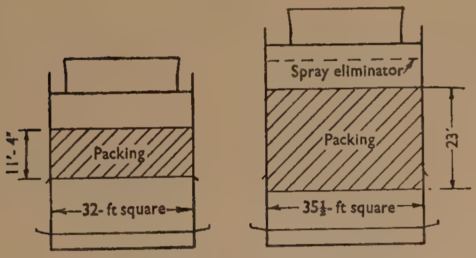


Fig. 18.—Characteristics of a splash-packing



Duty: 180,000 gal/hour 90°F to 72°F Wet-bulb 63°F

(a) FILM-FLOW TOWER (b) SPLASH-PACKED TOWER

Costs

Tower	Film-flow	Splash-packed
Initial capital cost of concrete shell, pond, fan, motor, etc.	£9,000	£10,700
Initial capital cost of packing	£4,950	£2,900
Annual cost of pumping water	£700	£930
Annual fan power cost	£900	£1,210

Fig. 19.—Comparative sizes of film-flow and splash-packed water-cooling towers

of water was to be in circulation, so that the manufacturers could decide whether it was better to install a film-flow or a splash-type tower; otherwise the purchaser might buy the wrong type of plant.

Mr R. I. Newsome (External Research Engineer, Messrs David Brown & Sons (Hudd.) Ltd) asked for additional information on gear-driven fans. There was an appreciable difference between the results obtained from tests on a gear unit alone and the figure of 75% given in Table 3. In order to measure the efficiency of the type of unit normally employed in a cooling-tower fan drive, a 5:1 ratio 6-in. centres worm-gear unit was interposed between a trunnion-mounted motor running at 1,140 r.p.m. and a water-cooled prong brake. Input and output torque values were recorded, from which the efficiency curve was calculated for full-rated power and smaller input powers. Even assuming an input power of 5 h.p. the efficiency was in the region of 90% and, since the no-load losses were constant and the tooth losses were directly proportional to the load, it was clear that the maximum efficiency was achieved only at full load, which in the unit in question was about 22 h.p. Some explanation of the difference between the two efficiency values was therefore needed.

The Authors touched on one explanation when they made the following comment on p. 96: "The lower efficiencies obtained in actual operation are thought to arise from out-of-balance aerodynamic forces, which result in extra friction in the bearings; this belief is supported by the unexpectedly high maintenance on bearings." Whether extra friction was present or not, there could be no doubt that bearing failures in standard vertical units operating under the conditions described in the Paper were above the normal expectation. That was more surprising when it was realized that all those units were operating well below their normal industrial ratings, and it emphasized the probability of the suggestion that the trouble was the result of some unknown extraneous condition.

His firm had developed a new type of gearbox for cooling-tower fan drives which embodied the following features. It had an extended bearing span, and extra roller bearings were fitted at the top and bottom of the wheel shaft in addition to the normal, angular contact thrust bearings. They had been fitted to cater for the out-of-balance aerodynamic forces or other unfavourable conditions occurring in operation. In addition, positive pumped lubrication was provided for in the top wheel-shaft bearings, a plunger pump supplying the oil. Double oil-seals separated by a grease chamber were fitted on each shaft projection. An umbrella-type of deflector was sometimes fitted at the output side of the unit to prevent ingress of water.

With regard to cost, the production of such units was of course non-standard, but the benefits to be gained from the use of that special type would, he believed, far outweigh the small extra cost involved.

Mr E. Talbot (Deputy Manager, Special Products Department, Head, Wrightson Processes Ltd) said that the Authors had reached some interesting conclusions on the design and operation of a particular type of mechanical draught cooling tower, but their conclusions could not be applied generally for the following reasons.

First, he felt that it was incorrect to draw an analogy between water cooling and gas absorption; although similar theoretical considerations might be applied, the practical problem was entirely different. Gas absorption usually involved scrubbing a gas with a clean liquid. Under such conditions a non-fouling film-type packing was ideal. In water-cooling, however, the packing of the cooling-tower was open to the atmosphere, and the gas—namely, air—contained all the dust and foreign matter associated with an industrial atmosphere. The water supply was very rarely totally enclosed, as usually the case with the liquid of a gas-absorption process, so that considerable water pollution, particularly from oily matter, could occur unless great care was exercised. Oil from such materials might cause the water film on the packing to break down into small rivulets, leading to splashing and excessive spray carry-over unless drift eliminators were fitted. Almost invariably in cost analyses drift eliminators were not included in film packings.

For those reasons, splash packings had been developed, because essentially they were non-clog packings ideally suited to industrial atmospheres.

Secondly, the statement at the foot of p. 102 that film-flow packing showed an advantage over modern types of splash tower was also open to doubt. Extensive experiments on an 18-ft-square test cooling-tower conducted in America on film-type packings had shown no advantage over the advanced type of splash packing currently used there. Those tests were substantiated because almost all American cooling-tower manufacturers used splash packing for their towers, which were between five and ten times more numerous than the total installed in Britain, including natural-draught cooling towers.

Thirdly, he could not agree with the conclusions on the mechanical equipment on cooling towers, with regard to the efficiency of gears decreasing substantially under cooling-tower conditions. Experiments had shown, on both small-scale and large-scale tests, that it was the fan which underwent a marked change in efficiency under cooling-tower conditions even when a streamlined throat to the fan stack was used. It had been found from tests on many types of fan that the fan efficiency was usually 10 to 20% less than for corresponding airflows when tested under British Standard conditions. Those results had been obtained on both direct-drive and spiral-bevel gear-drive units, and indicated that fan manufacturers' performance figures based on British Standard conditions were unreliable for cooling-tower applications, but gear manufacturers' efficiencies did not appear to be far wrong.

The unsatisfactory operation of reduction gears was a matter on which he endorsed Mr Newsome's remarks, but he did believe much of the trouble indicated in the Paper resulted from incorrect sizing of the boxes by the gear manufacturers. Adequate sizing of the gearbox was essential, since the box had to carry a large-diameter fan with its associated loads and vibrations. In Mr Talbot's experience, if the boxes were sized on the basis of the American Gear Manufacturers Association standards, with one or two extra refinements, they would be quite reliable. The main additions should be an increase in the spacing between the output-shaft bearings, the addition of extra roller bearings, and—a point which he thought had not been given sufficient attention—adequate stiffness of the gear case.

The remarks referring to American difficulties with mechanical equipment no longer applied, since first of all they had been based on a Paper[9] given in 1951. At that period cooling-tower manufacturers were very aware of the difficulties, and conducted extensive experimental tests to eliminate them. When Mr Talbot visited the United States in August 1955 he had been able to examine the operating records of more than 1,000 fan and gearbox installations on induced-draught cooling towers and, in contradistinction to what Mr Wood had said, there had been only one case in which serious trouble had arisen.

Mr R. F. Rish (Generation Design Branch, C.E.A.), congratulating the Authors on an excellent design for cooling the small amount of water required in chemical practice, said that when large quantities of water were concerned, on the scale of a central power station, if cooling was attempted by multiplying the number of small units very heavy capital costs would be entailed. For example, a circulation of 3,600,000 g.p.h., with a cooling range of 20°F and a 15° approach to a wet-bulb temperature of 62·8° and a humidity of 80%, would require one natural-draught tower about 330 ft high and 220 ft base diameter. It would cost about £160,000 ± £15,000. For mechanical-draught units using the methods shown by Jackson[4] there would be thirteen 32-ft-square cells. They would work out at £20,000 per cell, using the figures given in the Paper allowing for clearing the site and for extra pipes, cables, and switch-gear. The capital cost therefore would be about £260,000. When the power costs were taken into account, using 24-ft fans with direct drive, but making allowance for the smaller pumping head, the capitalized total cost of the mechanical-draught towers plus power costs came to 2½ times the cost of the natural-draught tower.

If a closer approach to the wet-bulb temperature than 15° was required, the position changed a little. If 11° was specified two natural-draught towers would be needed, each about 300 ft high and 190 ft diameter, at a cost of about £250,000. For the same duty, using grid-packed mechanical-draught cells, sixteen cells would be required at £20,000 each, making a total of £320,000. Taking the capitalized power costs into account on the same basis as laid down in the Paper, the total capitalized cost would then reduce to twice the cost of the natural-draught towers.

Those differential costs seemed to be rather higher than the Central Electricity Authority had experienced. When inquiries had been made for mechanical-draught towers and for natural-draught towers for the same duty, the capital costs had been about the same when piping and switchgear had been included. It was only when power costs had been taken into account that the mechanical-draught towers had to be rejected on economic grounds. The fantastically high cost for mechanical-draught towers appeared to be due to the attempt to multiply the scale by adding small units. If the capital cost was to be brought down, it seemed practical to build one large mechanical-draught unit with perhaps a 60-ft-dia. fan and a chimney 100 ft high. The chimney would take the plume away and prevent nuisance and recirculation, and would reduce the fan power by auxiliary natural draught. Such towers had already been built in Germany, and he had read of one at Essen built on that principle.

Finally, he would like to ask the Authors whether the duty chosen for the mechanical-draught cells had been proposed on the basis of engineering economics or had been laid down by the chemists of I.C.I. Ltd.

Mr F. F. Ross (Assistant to Chief Chemist, Central Electricity Authority) referred to the paragraph on timber life on p. 100. The Authors had stated that "soft rot" had been detected in one or two of their towers. Mr Ross said that the Central Electricity Authority had found "soft rot" in every tower which they had examined, with four exceptions. The exceptions were quite interesting. In one, the tower had been used for only 3 days a month, and had ample opportunity to dry out in between. In another the water was very heavily contaminated with iron salts downstream of a steelworks, whilst in another the health authorities required a certain quantity of chlorine to be kept in the water, and the chlorine at that concentration might possibly have protected the timber. In the fourth case the tower had been in service for only about 6 months. He had not examined it recently, and it might now be suffering from "soft rot."

He did not want to cause alarm and pointed out that "soft rot" was very slow. Under average conditions it could be reckoned to eat away the wood from the surface at about 0·01 in. per year, which meant that the life of unprotected timber under average conditions largely depended upon its shape and thickness. In that respect Mr Ross agreed with Mr Chilton, who thought that ⅜-in. timber would not last so long as timber substantially thicker. Mr Ross had seen timber of triangular shape, cut diagonally from 2-in. × 2-in. timber, which had been in service for more than 25 years. It had rotted about ¼ in. in from each surface, but the rotted wood was still in its original shape and fully effective. That was unusual, and in the ordinary way the wood rotted and flaked off at a rate which was probably affected by the type of water, the degree of recirculation, and the load schedule. Very little was known about that.

The Authors had said that preservatives had not ensured complete immunity and that further trials were being made with a new preservative. Trials were also being made with an old preservative, but at a greater concentration than before. The present position was that there were two water-borne preservatives which contained salts toxic to "soft rot" fungi. Their toxicity had been proved in the laboratory and in short-period (one-year) tests in cooling towers. By analysing wood which had been in service for many years it had been shown that those particular salts remained fixed in it. If used in sufficient strength to start with, it was therefore reasonable to expect that they would remain effective. Since 15 years was not an economic life for power-plant auxiliaries the C.E.A. were trying the preservatives at an adequate level.

Mr G. C. Parkinson (Chief Engineer, Cooling Tower Dept, L. G. Mouchel & Partners Ltd) said he had been interested to read in the Paper of a development parallel to the system expanded by Hutchison and Spivey[2] in 1942. Mr Parkinson's firm had been concerned with that development and had found that the performance per cubic foot of packing was exactly similar. Wing-Commander Cave-Browne-Cave had asked about the timber boards. The Hutchison and Spivey packing had 5-in.-deep boards at 1¾-in. centres, and the performance was obtained with two-thirds of the horsepower required by the ⅜-in. board packing. The Hutchison and Spivey Paper also showed that performance with deep boards of the type described as used in Germany was ineffective compared with boards at right-angles. The performance was greatly stepped up when the 5-in. boards were turned at right-angles in alternate layers.

Fig. 4 showed a 2- to 4-in. gap in film-flow packing. That could cause splash droplets because the water had to drop from the top fixed packing to the lower packing.

It was known from experience that packing with any type of film flow if resting on top of itself tended to compress. His firm was at the moment concerned with the first re-packing of film-flow packing which was planned to take place in 1956, 11 years after installation. That packing was compressed with a loss of approximately 1 ft of height on 11 ft of packing.

The Authors had mentioned that vegetation growth in mechanical-draught towers had not been experienced. The tower to which Mr Parkinson referred was coated with the same sort of film found in natural-draught towers, and where it fell into the troughs it could cause an enormous amount of trouble.

Where the water was dirty, film-flow packing should not be used. In one instance in a tower connected with gas purification the guaranteed approach to the web-bulb temperature was 8°F. After 2 years the top of the louvers had become covered with a film, which dropped the water straight down through the packing. The approach to the wet-bulb temperature then became 25°F instead of 8°F. The original back pressure had been 0·26-in. (water-gauge), but the actual back pressure became 0·44-in. w.g.

In America, reversible fans were used for the prevention of ice formation. Could the Authors say anything about ice formation on the packing of film-flow towers?

The Authors, in reply, referred to Wing-Commander Cave-Browne-Cave's question about the use of vertical sheets instead of the criss-cross grid arrangement. The Authors had chosen the shallow grid, in which the depth of the grid was of the order of the pitch of the grid, because the effectiveness of the surface in the shallow grid was very much greater than that of the plain vertical sheet. If the plain vertical sheet were given a figure of performance of 1, then the type of grid which they had used had a surface effectiveness of the order of 4. At the same time the pressure drop per foot of packing also increased by a factor of about 16. Thus it was necessary to weigh the advantages of lower air-pressure drop for vertical sheets against the advantages of less surface and lower water-pumping height for grids. The Authors' calculations had favoured grids. However, it was known that vertical sheets had been used in a natural-draught tower in the United Kingdom and were widely used in mechanical-draught towers in Germany. Mr Parkinson had said that he used grids where the depth was of the order of 5 in., instead of the 2 in. used by the Authors and, whilst supporting the advantages of grids over vertical sheets, he considered that the deeper grids were superior in that they gave a similar transfer performance at two-thirds the cost in air-pressure drop. It was known that those deeper grids (although not stated by Mr Parkinson) were ¼ in. thick, and in the light of Mr Ross's data on soft-rot attack it was clear that a ⅜-in.-thick lath would have a longer life. If, for instance, it was assumed that the rate of decay was 0·01 in/year (both sides) and the lath was renewed when the thickness of good timber had decreased to ⅛ in., the length of life would be 6 and 12 years respectively. So again it was necessary to strike an economic balance between thin laths and low pressure-drop as against thicker laths and longer life. A fully effective timber preservative would make possible the use of thin laths and the Authors, in agreement with Mr Ross, attached much importance to the development of the "perfect" timber preservative.

A number of speakers had referred to the low transmission efficiencies on the gear-driven fans. Subsequent to the preparation of the Paper the Authors had carried out experiments in which thermocouples were fixed to a gearbox, and the rise of temperature noted from start-up. The slope of the curve gave the amount of heat generated in the gearbox, which corresponded to an efficiency of the order of 78%, compared with the average figure of 75% for the 12-ft-fan gearboxes. That seemed to prove conclusively that friction in the gearbox was greater than expected from the manufacturer's specification. The development described by Mr Newsome of a new type of gearbox for cooling-tower fan drives was therefore of particular interest.

It was possible that Wing-Commander Cave-Browne-Cave's suggestion of carrying the fan on a fairly long shaft might provide a cheaper solution in allowing a standard normally rated gearbox to be used.

With regard to Mr Chilton's questions about recirculation when the wind direction was along the axis of the tower, the Authors stated that they had no exact information but agreed that there might well be an appreciable amount of recirculation when the tower was very long. It should be emphasized that mechanical-draught towers were used mainly for smaller applications where recirculation was no problem. The recirculation ratio of 8% referred to an infinitely long tower in a cross-wind under the most unfavourable wind conditions. The average over the year on an existing large installation was estimated to be 4%, and it was hoped that the improved baffle arrangements referred to in the Paper would reduce that to 2%. The Authors favoured the central baffle because it was quite effective in preventing loss of water at the base of the tower and did not constitute a resistance to the incoming air.

On the question of splash packing *versus* grid packing, the Authors agreed with the comments of Messrs Ripley and Hutchinson, which answered some of the points raised by Messrs Chilton and Talbot. It would seem that no one packing was the universal solution and that the cooling-tower manufacturer should be in a position to offer splash or film-flow packing according to which suited the customer's requirements best. Although not suitable for very dirty conditions, as pointed out by a number of people, particularly where there were algae growths, grids were originally developed in the early nineteen-thirties as a non-clog packing for use in flue-gas scrubbers on P.F. boilers and on dirty process gases in gas works. A number of the Authors' cooling towers took water which had been in direct contact with compressor gases. One would expect, as mentioned by Mr Talbot, that oily deposits would break the water films. That was not so and the towers were operating quite satisfactorily.

Most of the comments on fans were made by Mr King-Smith, and answers to his queries were briefly that no icing trouble had been encountered on fans in the induced-draught position, even when not in operation; a very thin layer of dirt, say $\frac{1}{64}$ in. thick, accounted for the 4 to 5% reduction in fan efficiency; the static pressure rise across the fan was measured by a sensitive manometer with multi-point leads to below the fan housing (inside the tower) and to the tower roof; the slight departure of fan performance from specification was partly reflected in higher power consumption and partly in a lower air rate, and finally four-bladed fans were used instead of two-bladed fans because the manufacturer's efficiency figures showed that the higher initial cost of the four-bladed fans was justified by its higher efficiency. The Authors wished to point out that the high efficiencies given in Fig. 7 of the Paper for the 24-ft-dia. fans were based on a limited number of tests and recent check figures were more in keeping with the manufacturer's characteristic curve. The higher efficiency of the 24-ft, compared with the 12-ft, fans might well be, as Mr King-Smith had suggested, due to the radiused housing; this point would shortly be examined by tests on a 12-ft-fan housing with a radiused inlet.

Messrs Rish and Wood had stated that for power-station conditions natural-draught towers were preferable to mechanical-draught. The Authors would agree but found that for the lower water-circulation rates and close approaches to the wet-bulb temperature common in process industries (single installations seldom exceeded 1,000,000 gal/hr) mechanical-draught was more economic. Incidentally, Mr Rish's cost data overstated the case against mechanical-draught. The basic, i.e., contractor's cost of a 32-ft sq. unit was nearer £15,000 than £20,000 and might well be less for a very large installation. The figures given in the Paper were realistic achieved costs and contained many items not normally included by the contractors.

Finally, with regard to the merits of forced and induced draught, referred to by Mr Wood, the Authors maintained that induced draught was essential to reduce the effects of recirculation in the large installations. Efforts must, therefore, be made to eliminate the present imperfections of fan-drives. When a reliable induced drive had been developed the Authors considered that it would more than hold its own against forced draught on the grounds of freedom from icing trouble, better air distribution, and less dirt deposit on the fan blades.

The closing date for Correspondence on the foregoing Paper has now passed without the receipt of any communication.—Sec.

ORDINARY MEETING

13 December, 1955

WILLIAM KELLY WALLACE, C.B.E., President, in the Chair

The Council reported that they had recently transferred to the class of

Members

Brand, Norman Pumfrett, B.Sc.(Eng.) (*London*).
Britton, Charlie Alistair, M.Eng. (*Liverpool*).
Buchanan, Robert Cunningham, *M.B.E.*, B.Sc. (*Glasgow*).
Buckley, Dennis Harold, B.Sc.(Eng.) (*London*).
Durrant, Theodore Vincent Scott, M.A. (*Cantab.*).
Holloway, Brian George Ronald.
Houston, William Mackay St Clair, B.Sc. (*Glasgow*).
Jessop, Charles Uttley.
Keir, Falconer, *M.M.*, B.Sc. (*Edinburgh*).
Lewis, John Addison, *M.B.E.*, B.A. (*Cantab.*).
Mehra, Sita Ram.
Morrison, William, B.Sc. (*Edinburgh*).
O'Donnell, Hugh, B.Sc. (*Edinburgh*).
Rose, Horace Edgar, D.Sc. (*London*), Ph.D. (*Manchester*).
Smith, Harold John.
Sparks, Cedric Harold, M.A. (*Cantab.*).
Thomas, Derek George.
Winston, William Peter, B.Sc.Tech. (*Manchester*).

and had admitted as

Graduates

Agnew, Roderick, B.Sc. (*Belfast*), Stud. I.C.E.
Alford, Philip Leonard, Stud.I.C.E.
Amstad, Roy Jonas.
Angwin, John Osborne, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Bailey, James Leslie.
Barcroft, William Arnold, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Bartley, Bryan Allen, B.E. (*New Zealand*), Stud.I.C.E.
Beaver, Brian John, Stud.I.C.E.
Bloomfield, John Alfred.
Bowden, Malcolm Henry Charles, Stud. I.C.E.
Brand, Robert James, B.Sc. (*St Andrews*).
Breakey, John Macdonald, B.Sc. (*Belfast*), Stud.I.C.E.
Breakspear, Michael John, Stud.I.C.E.
Brest, Stanley Herman, B.Sc. (*Cape Town*).
Brown, William McNaught, Stud.I.C.E.
Canavan, Michael Dermot, B.E. (*National*).
Chambers, John Arthur.
Cheek, Grahame Noel, Stud.I.C.E.
Clayden, Kenneth John Charles, B.Sc. (*Leeds*), Stud.I.C.E.
Coates, Malcolm, B.Eng. (*Liverpool*).
Collier, John, Stud.I.C.E.
Constable, Maurice Banbury, B.Sc. (Eng.) (*London*).
Cooper, Roger Parnell, Stud.I.C.E.
Couzens, Michael Leonard, Stud.I.C.E.
Cowley, Thomas, B.Sc.Tech. (*Manchester*), Stud.I.C.E.
Cunliffe, James Walter Frederick, Stud.I.C.E.
Darbyshire, Charles Philip, B.Sc.Tech. (*Manchester*).
Davison, Christopher Ernest, B.Sc. (Eng.) (*London*), Stud.I.C.E.
Dean, Peter, B.Sc.(Eng.) (*London*), Stud. I.C.E.
Doidge, John Wall, Stud.I.C.E.
Dunn, Nicholas Howard, B.A. (*Cantab.*), Stud.I.C.E.
Dymond, Edward John, Stud.I.C.E.
Ganesan, Mahesan, B.Sc. (*Ceylon*), Stud. I.C.E.
Grandorge, Richard Britton, Stud. I.C.E.
Harmsworth, Leslie George.
Hartnell, Raymond, B.Sc. (*Bristol*), Stud. I.C.E.
Haynes, John Carleton Douglas, B.Sc. (*Bristol*), Stud.I.C.E.

Hayworth, Alexander Liddell, Stud.I.C.E.
Henderson, Ralph Macnicol, Stud.I.C.E.
Hicks, Arthur Blamey, B.C.E. (*Melbourne*).
Hoctor, Michael John Augustine, B.E. (*National*).
Howe, Kenneth, B.Sc. (*Manchester*), Stud.I.C.E.
Huang Chung Ling, B.Sc. (*Hong Kong*).
Jeyarajah, Reginald Rajakone, B.Sc. (Eng.) (*London*), Stud.I.C.E.
Jolliffe, Harry Raymond, B.Sc. (*Bristol*), Stud.I.C.E.
Kirk, Thomas Blackwood, Stud.I.C.E.
Lawrence, Howard Trevor, B.Sc.Tech. (*Manchester*), Stud.I.C.E.
Lobban, William George, B.Sc. (*Aberdeen*), Stud.I.C.E.
McBean, Peter James Alexander, Stud.I.C.E.
McCracken, Victor Hugo, B.Sc. (*Belfast*), Stud.I.C.E.
McDermont, John Lawrence, B.Sc. (Eng.) (*London*).
McDowall, Andrew Douglas, Stud.I.C.E.
Majid, Kamal Izaldeen, B.Sc. (*Manchester*).
Mansell, Edward Russel Guy, B.Eng. (*Sheffield*).
Mayfield, Colin, Stud.I.C.E.
Merrett, Glyn, B.Sc. (*Bristol*).
Murray, John.
O'Neile, Murray Robert, B.Sc.(Eng.) (*London*).
Overend, Philip Robinson.
Pankhurst, Richard Norton Waller, B.Sc.(Eng.) (*London*).
Parrott, Allan Peter Willson, Stud.I.C.E.
Powell, David Arthur, B.Sc. (Eng.) (*London*), Stud.I.C.E.
Quick, Michael Clifford, B.Sc. (*Bristol*).
Redshaw, Victor James Waddell, Stud.I.C.E.
Reith, Archibald, Stud.I.C.E.
Robinson, Miles Trevor, B.A. (*Oxon.*).
Robinson, Peter John, B.Sc.(Eng.) (*London*).
Rowe, Lawrence Edward, B.Sc.(Eng.) (*London*).
Rutherford, James, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Saines, Alan John, Stud.I.C.E.
Schaeffer, Theodore John, B.Sc. (Eng.) (*London*), Stud.I.C.E.
Sharp, John Stephen George, B.Sc.Tech. (*Manchester*), Stud.I.C.E.
Shoreson, Wilfred, B.Sc. (*Leeds*), Stud.I.C.E.
Smith, Malcolm John Rognvald, B.Sc. (*Aberdeen*).
South, Ronald James, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Steel, Dudley John, Stud.I.C.E.
Thorn, John Haddon, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Tietz, Stefan Berthold, B.Sc.(Eng.) (*London*).
Tildersley, John William.
Tolefe, Augustine Uchechukwuka, B.Sc. (*Southampton*).
Turner, Charles Peter Owen, B.E. (*New Zealand*), Stud.I.C.E.
Tyrer, John Edward, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Urquhart, William Ross Fraser, B.A. (*Cantab.*), Stud.I.C.E.
Vickery, Kenneth William, B.Sc. (*St Andrews*).
Walls, Benjamin John Stuart, B.Sc. (Eng.) (*London*), Stud.I.C.E.
Wilkinson, Christopher John, B.Sc. (Eng.) (*London*), Stud.I.C.E.
Williams, Henry Browne, B.Sc. (*Belfast*), Stud.I.C.E.
Wright, Roy, B.Sc. (*Nottingham*).
Wykes, Robert Henry Francis Paul, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Yalcin, Yilmaz Orhan, B.Sc. (*Durham*).

and had admitted as

Students

Adams, Peter Derek.
Adamson, Alexander Stuart.
Alderson, Peter James Alan.
Ashcroft, David Julian.
Atkinson, Richard.
Azmeh, Mohammad Mumtaz.
Barker, John Raymond.
Barrett, Alan John.
Bateman, Colin Ian.
Batten, Roger Maxwell.
Bommer, Christopher Maria.
Brindley, Charles Stuart.
Brown, Robert.
Browne, Roger Coulton.
Bullock, Frank Maxwell.
Cameron, John Barry Chibnall.
Castle, Robin Leslie.
Chalk, Edward Charles Patrick.

Chaudhuri, Deb Kumar.
Clark, Brian Edward.
Clarke, Michael Rushton.
Cobb, Philip Gibson.
Corin, John Desmond.
Davis, Colin Peter.
Dawson, David Browell.
De Lord, Bernard Pereira.
Doncaster, Alan Arthur.
Edmiston, James Given.
Ellwood, John Frank.
Erskine, Thomas Adrian.
Evans, Brian Wynn.
Everett, Neville Stewart.
Fairall, Keith Ian.
Ffitch, Dundas Harold.
Ford, William.
Fulthorpe, Carlton James.
Gadsby, John William.
Gandy, Alistair Anthony Kendall.
Gilliespie, Noel Maxwell.
Graham, John David.
Gray, John Eric.
Griffiths, Michael Campbell.
Hancock, John William.
Harrison, William Jerome.
Hawkeswood, Barry.
Herring, Kenneth George Albert.
Hewines, Robert Frederick.
Hoadley, Anthony David.
Hope, Brian Bradshaw.
Hunter, Hugh.
Hussein, Haji Mohamed.
Iley, Peter.
Jarvis, John Roger.
Kennedy, Donald Hugh.
Kennedy, Francis Brian.
Kennedy, Peter Charles John.
Ladchumanan, Kandiah.
Leishman, Stanley Henry Michael.
Lewis, Michael George.
Lewis, Peter Govey.
Lindsay, Ernest Dennis.
Lipton, Allan.
Lytle, Samuel James.
McDonald, Alan.
Marshall, John David.
Maxwell, William Hall Christie.
Mears, Colin Charles.
Millea, Trevor Graham.
Millen, Anthony David.
Miller, David Lewis.
Monks, Robert.
Muir, Hugh.
Munyard, Austin Fairley.
Myerscough, Cedric Thomas.
Nicholas, Llewelyn Sidney.
Paramjothy, Sundarampillay.
Pargiter, Malcolm Charles.
Parkin, John Waterhouse.
Partington, Derek.
Patel, Jayanti.
Patterson, William Adrian.
Peacock, Martyn Richard.
Perrett, Stephen John.
Piall, George Henry Anthony.
Potter, Brian Albert.
Reynolds, Patrick James.
Richards, David Mills.
Rishard, Izzudheen Mohamed.
Ruthirapathy, Kanthasamy.
Sanderson, Colin.
Shearer, Douglas Murray.
Smith, Michael Andrew.
Smith, Rodney Charles.
Stevens, Kenneth.
Summerfield, Frank.
Summerton, John David.
Teng Matthew Law Kin.
Thomas, Daniel.
Thompson, David Harloe.
Thompson, Ralph Graham.
Thompson, Roger.
Trill, John Ernest.
Trippner, George.
Trotter, William Maurice.
Turner, Frederick.
Vincent, Arthur Westcott.
Vine, Peter Raymond.
Wah Dennis Fong.
Wainwright, David Alan.
Watson, Brian Stuart.
Watson-Jones, Robert Donald.
Watts, Ian Lea.
Webster, Peter Thomas.
Welch, Martin Stuart.
Whyte, Marshall William.
Williams, Robert Stephen.
Williams, Thomas Harold.
Winton, John Hubert.
Wise, Malcolm.
Wood, John William.
Wydenbach, John Clive.
Yeadon, Eric.
Yong Yung Ta.
Young, Campbell Millar.

The President said the Parsons Memorial Lecture would be delivered by Sir Harold Roxbee Cox. The management of the Lectures was in the hands of the Royal Society and the Institution was one of the bodies asked from time to time to hold one of them. This was the twentieth Memorial Lecture and the third to be arranged by the Institution.

Sir Harold Roxbee Cox then delivered his Lecture.

THE SIR CHARLES PARSONS MEMORIAL LECTURE, 1955

THE DEVELOPMENT OF THE GAS TURBINE

by

Sir Harold Roxbee Cox, D.Sc., Ph.D., M.I.Mech.E., F.R.Ae.S.

INTRODUCTION

When I was asked to deliver the Parsons Memorial Lecture on the development of the gas turbine I was conscious of two reactions in quick succession. The first was a feeling of pleasure—I was greatly honoured by the invitation. The second was a feeling of disquiet—had not so much been said about the gas turbine, not least by myself, that there remained little, especially for one who had long ago left the ranks of the scientists and designers for the harsher realms of administration, for me to say? Whilst this disquiet remained for a long time—it is, indeed, not yet wholly dispelled—I was persuaded that there was room for a review which put the gas turbine in historical perspective, which was sufficiently technical in content to qualify

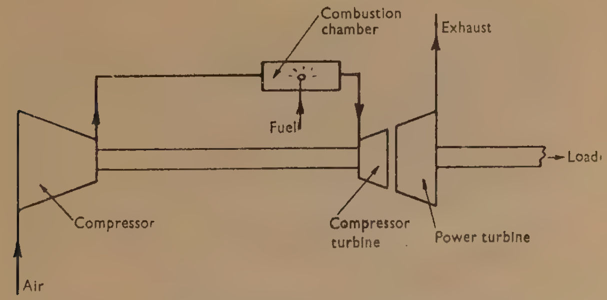


FIG. 1.—SIMPLE INTERNAL-COMBUSTION OPEN CYCLE

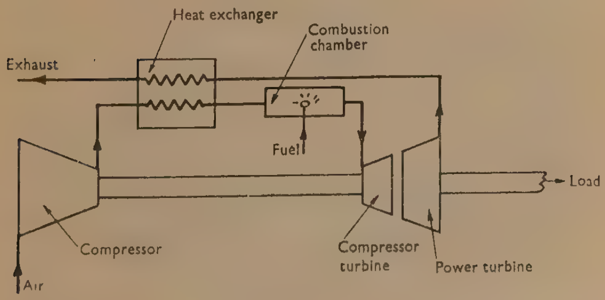


FIG. 2.—INTERNAL-COMBUSTION OPEN CYCLE WITH HEAT EXCHANGER

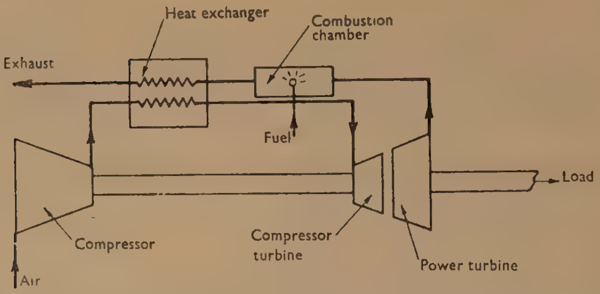


FIG. 3.—EXTERNAL COMBUSTION OPEN CYCLE

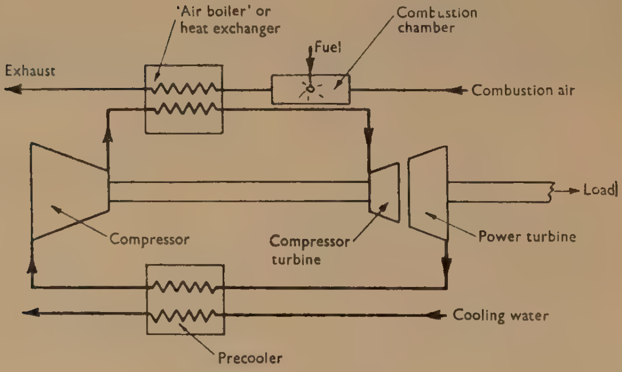


FIG. 4.—CLOSED CYCLE

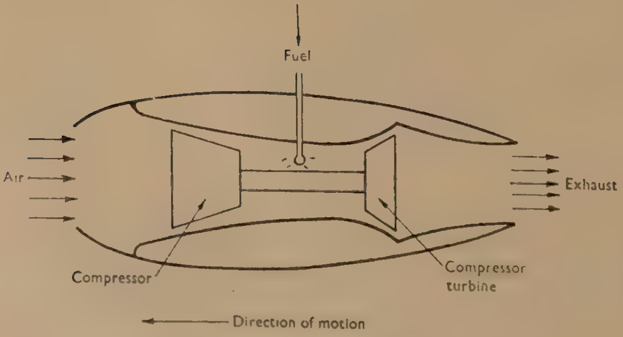


FIG. 5.—JET-PROPULSION ENGINE (TURBO-JET)

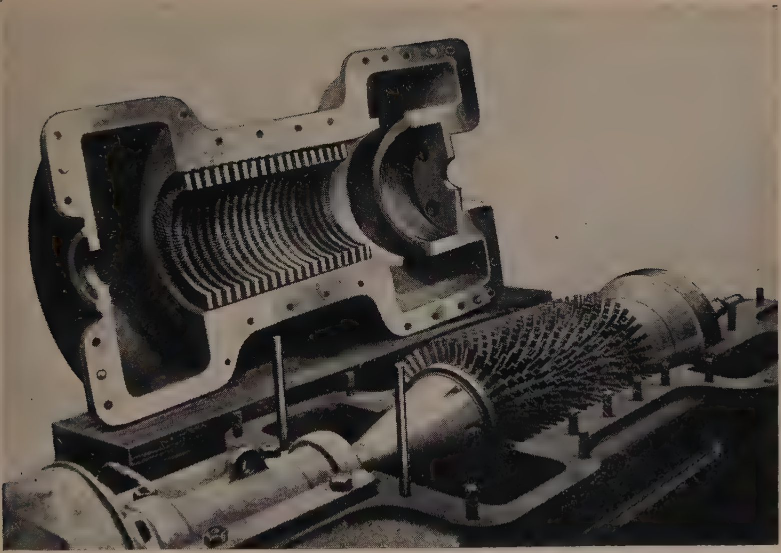


FIG. 12.—PARSONS'S FIRST COMMERCIAL AXIAL COMPRESSOR

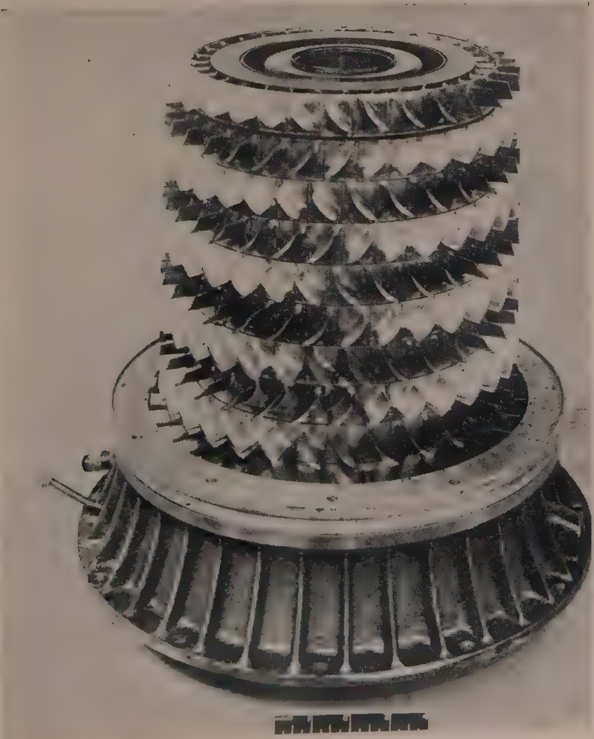


FIG. 15.—GRIFFITH CONTRA-FLOW CONTRA-ROTATION TURBO-COMPRESSOR, BUILT BY ARMSTRONG SIDDELEY, 1939, TESTED AT R.A.E., 1940

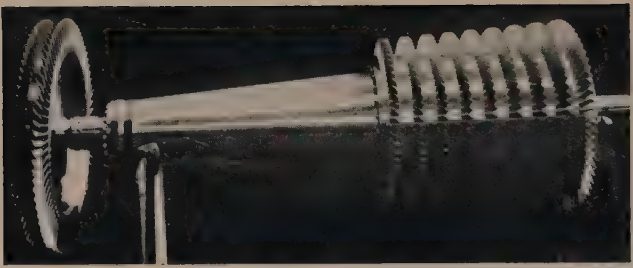


Fig. 16.—Rotor of the Metropolitan-Vickers axial-compressor F.2 jet-propulsion engine

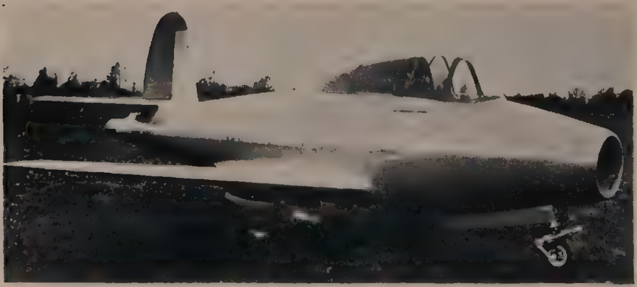


Fig. 18.—The first British jet-propelled aeroplane, E.28/39

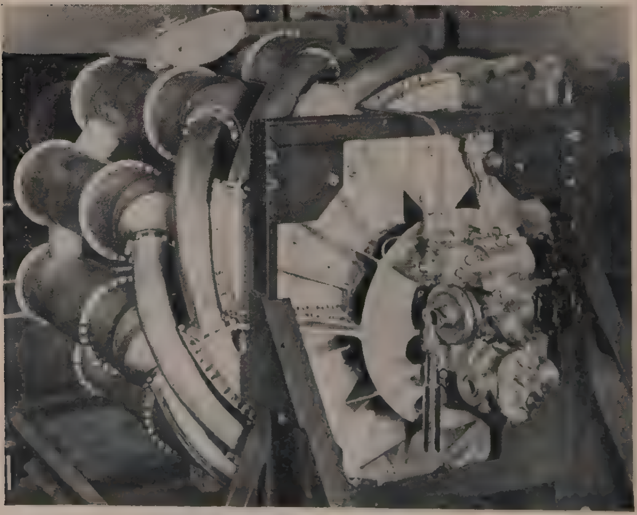


Fig. 19.—The Whittle W.1 engine of the Gloster E.28/39

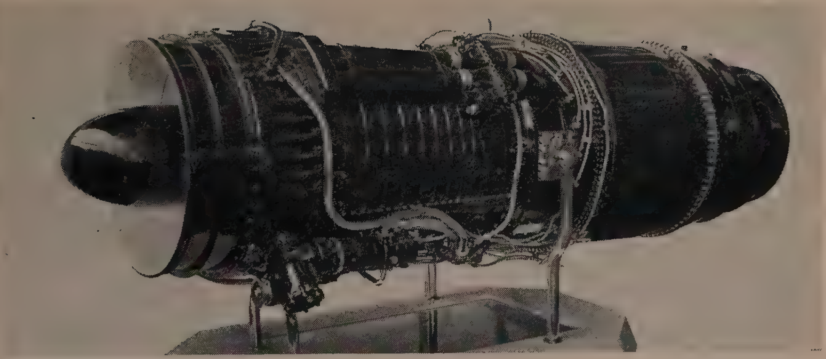


Fig. 24.—The Armstrong Siddeley "Sapphire"

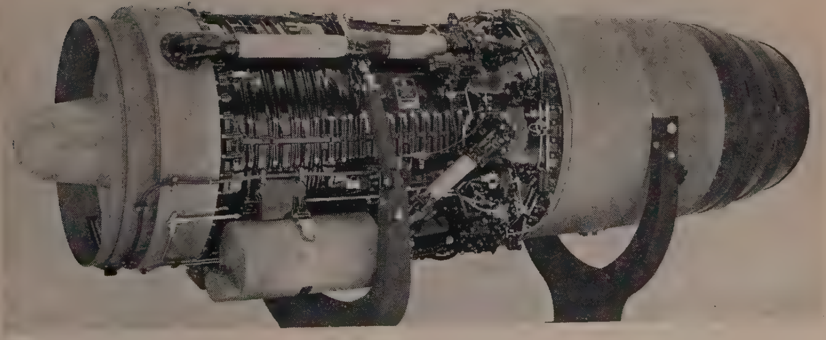


Fig. 25.—The Rolls-Royce "Avon"

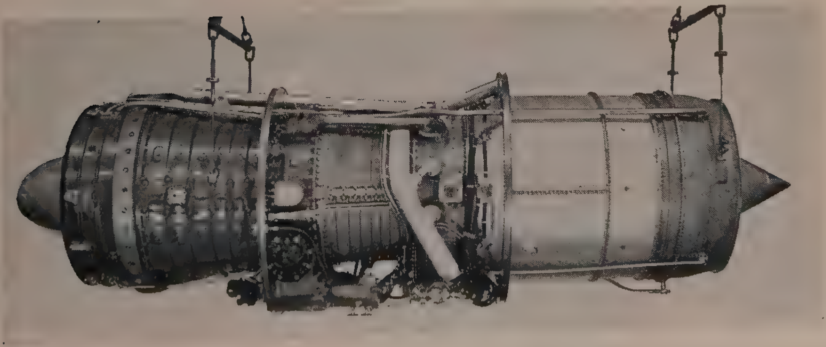


Fig. 26.—The Bristol "Olympus"

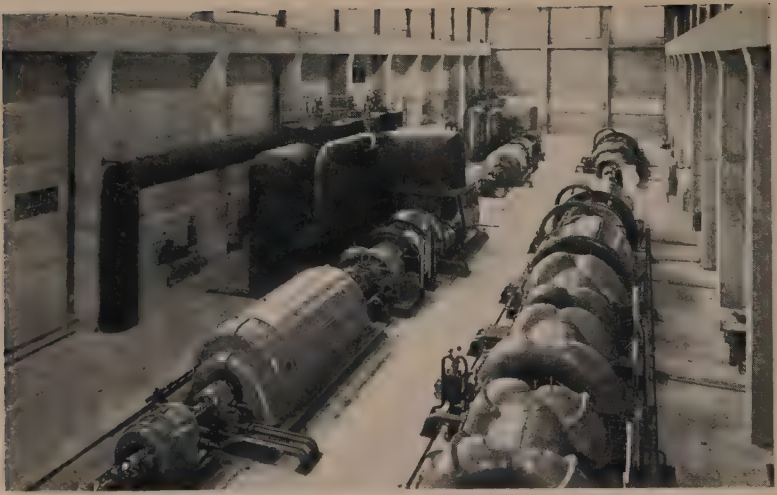


Fig. 29.—The 40,000-kW Brown Boveri plant at Beznau

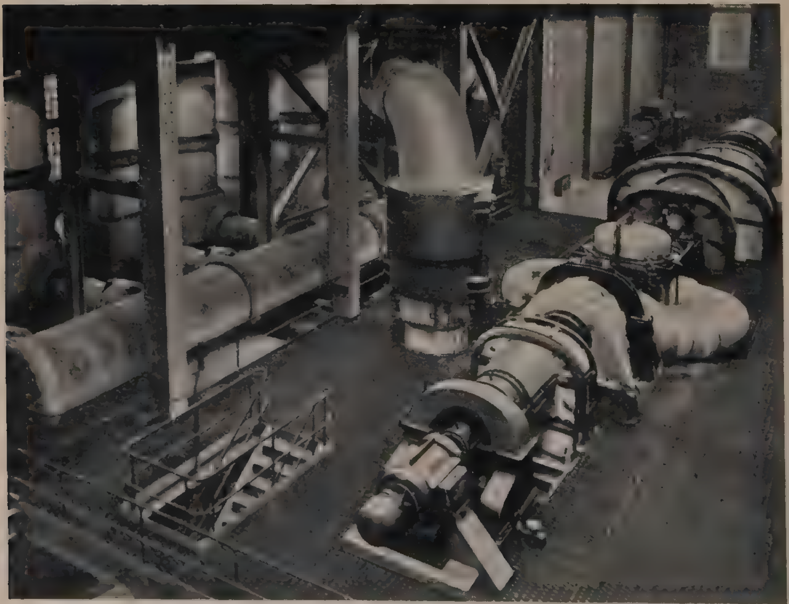


Fig. 30.—The 15,000-kW Metropolitan-Vickers plant at Stretford power station

for the annals of a learned society of engineers, and which was insufficiently so to be a bore when read *in extenso*. Despite the rigour of the last part of that specification, I decided therefore to accept the honour offered and it is indeed a pleasure to be reading to-night this lecture in memory of an engineer than whom there has been none greater—Sir Charles Parsons.

Heat engines are of two kinds—those in which the working fluid is heated by means external to itself, and those in which the working fluid is heated by combustion within itself.

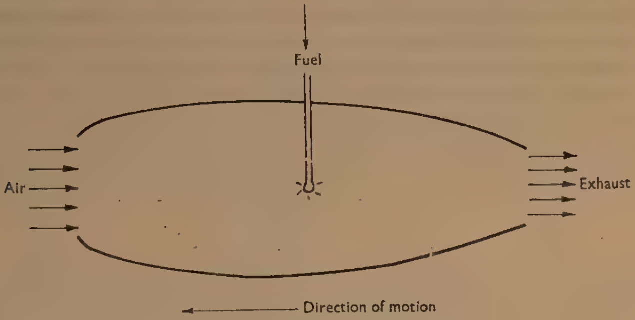


Fig. 6.—Ram jet

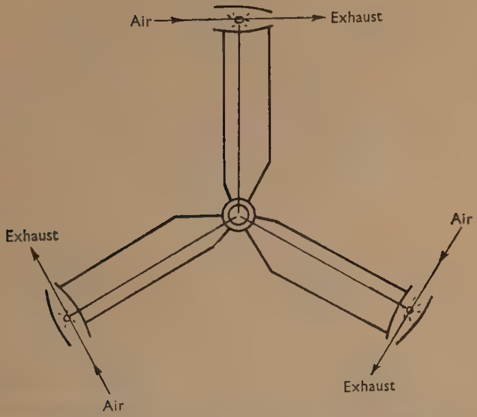


Fig. 7.—Principle of ram-jet helicopter

In the realms of practical power plants, there have for long been piston engines in both classes but turbine engines only in one. In recent years, however, we have seen the practical development of the internal-combustion turbine, so there are now turbine engines in both classes and the symmetry is complete. It is difficult to visualize any practical means of converting heat energy into mechanical energy except by engines of these four kinds, or combinations of them or their parts. With the development of the gas turbine therefore we may consider, if we so wish—and I, for one, confess to finding the thought attractive—that a pattern in the great picture of energy transmutation has been completed. We can elaborate the pattern, we can fill in the detail, but we cannot enlarge it. There would appear to be no other practical engineering ways of driving shafts by means of heated fluids, though the heat of nuclear fission can be substituted for the heat of chemical combustion. There are attractive hybrids—the gas turbine employing external combustion, called the closed-cycle gas turbine, and the Pescara machine in which the compression is performed by pistons and the expansion is through a gas turbine rotor. But for prime movers different in essential character we must await the development of the so-called fuel-cell, in which chemical reactions are transformed into electrical energy without the intervention of machinery. More remote, but conceivable, is the similarly direct transformation of atomic energy into electrical energy. For a long time to come, however, the practical means of transforming it will be by steam or gas turbines.

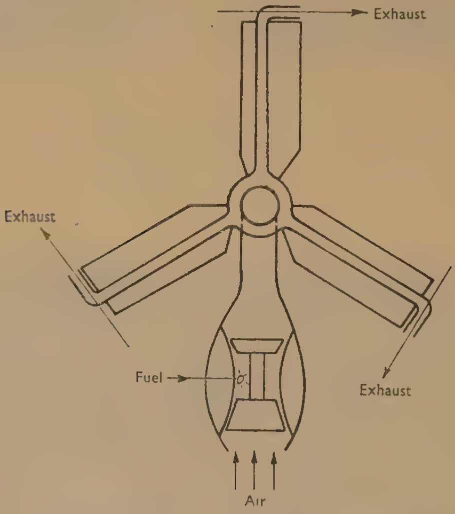


Fig. 8.—Principle of turbo-jet helicopter

The English language grows in a regrettably haphazard way and in the newer fields of engineering, wherein we might expect a precision in nomenclature comparable with the precision demanded in manufacture, we find instead generous tolerances which James Watt would not have admitted to his vocabulary, whatever he might have felt appropriate to his engines. The words "gas turbine" are themselves ambiguous. Do they refer to a complete engine or to the turbine rotor in it? The word "burner" is habitually used in the United Kingdom for the fuel injector nozzle and not, as would seem more appropriate, for the combustion chamber. Generally, by "gas turbine," I shall mean "gas-turbine engine," the machine which in its simplest form consists of a compressor, delivering air to a combustion chamber in which it is burned with fuel and from which it passes through a turbine system which drives the compressor and a power shaft (Fig. 1).

Application of the gas turbine in aircraft has produced special variations. The conventional form of gas turbine designed to drive an airscrew conforms to my brief definition, but in the simple jet-propulsion engine (Fig. 5), the turbine system drives only the compressor and the maximum possible amount of energy is left in the issuing jet, the reaction from which directly produces forward motion.

Forward speed itself induces compression. Produced in this way, the compression is called "ram." At very high speeds the ram in a jet-propulsion engine can be so great that a mechanical compressor is superfluous. The need for a turbine wheel consequently disappears, and the engine becomes merely an appropriately shaped pipe in which fuel is burned (Fig. 6). This engine, which is called a ram jet, is clearly not a gas-turbine engine. But if we put ram jets on the periphery of a wheel reacting tangentially to it we would appear to have another form of gas turbine (Fig. 7). A variant of this is a jet-propulsion engine delivering its reaction tangentially at the rotor tips (Fig. 8). In this form, we seem almost to have slipped back 2,000 years to Hero (sometimes called Heron) of Alexandria, father—indeed, grandfather—of engineers; but the arrangements, which I think we must class as internal-combustion turbines, have been successfully used for driving helicopter rotors.

EARLY HISTORY

The commonest form of motion man requires of an engine is rotational motion and it is perhaps a little odd that turbine heat engines were not developed sooner. Not only did windmills and water wheels set obvious examples, but the first heat engine idea of which we have record, Hero's (Fig. 9), proposes the direct generation of rotation. The need to lift water from mines, however, quite naturally produced the idea of piston engines, and their successful development reasonably led to their adaptation to the task of generating rotation. But without a pumping problem, development might have proceeded differently, for intelligences which refused to be bounded by convention existed very early in modern engineering history. Among them John Barber's is worthy of more than passing mention.

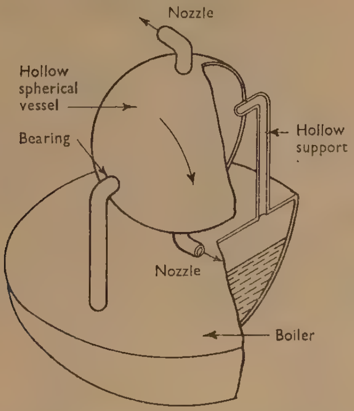


FIG 9.—DIAGRAM OF THE ENGINE OF HERO (OR HERON) OF ALEXANDRIA (*circa* 150 B.C.)

Barber seems to have been a turbine-minded man in a piston age. Bruce has studied his patents[1] and refers to "a form of water turbine" in a pumping engine patent of 1766, and an elementary form of impulse steam turbine in a patent of 1776. The gas turbine patent for which we remember him today was No. 1833 of 1791 (Fig. 10). The text of the patent, less the formal preamble and ending, is quoted in Appendix I and shows that Barber not only described a gas turbine, but proposed water injection for augmenting "both in quantity and velocity" the fluid stream—a device used in recent years for boosting jet thrust for aircraft take-off. Incredibly,

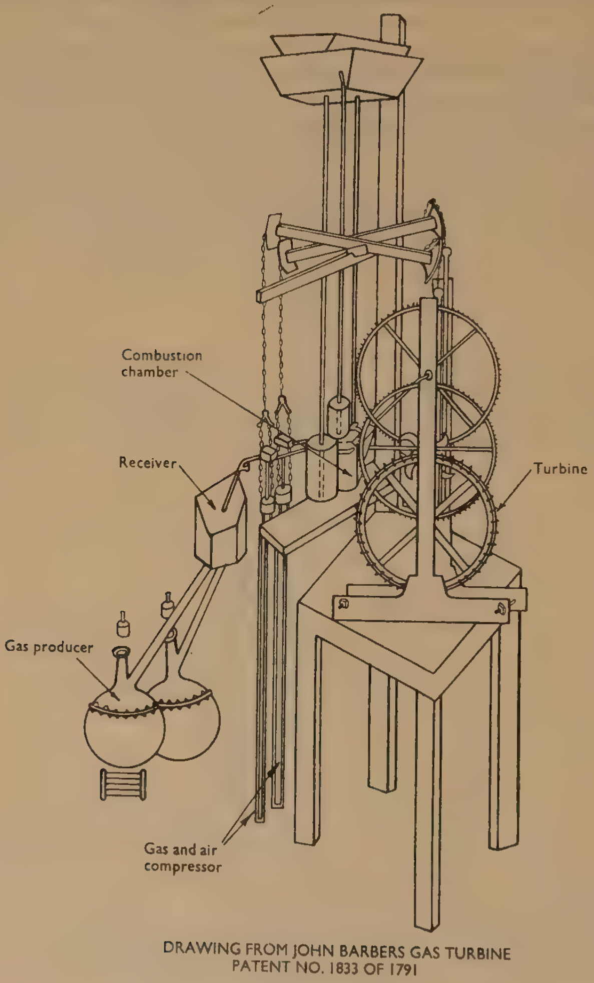


FIG. 10.—DRAWING FROM JOHN BARBER'S GAS TURBINE PATENT NO. 1833 OF 1791

in addition, Barber referred to passing the fluid stream "out at the stern of any ship, boat, barge or other vessel so as by an opposing and impelling power directed against the water carrying such vessel, the vessel with the contents may be driven in any direction whatsoever." In so proposing to use a gas turbine for jet propulsion of a water vessel Barber came so near to anticipating Whittle that one feels that had he met his distinguished contemporary, Sir George Cayley, the "father of British aeronautics," he might have done so—on paper, at any rate. Even though Barber's gas turbine did not get far beyond the paper stage—and there is no evidence that it did, although Bruce finds good reasons for suggesting that he made practical experiments—it is a remarkable proposal and one for which his name will forever be revered.

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[1] The references are given on p. 156.

Almost a century after John Barber's patent there appeared another in which was recorded what Sir Claude Gibb and Dr Bowden have rightly called "the first rational conception of a gas turbine as we today interpret the name."[2] This was the patent specification No. 6735 of 1884 in which the engineer we are honouring this evening, Sir Charles Parsons, described a gas turbine in terms similar to those I used earlier this evening. I have included the relevant part of the specification in Appendix II, from which it will be seen that Sir Charles proposed the cooling of the turbine blades, a refinement essential to the achievement of high thermal efficiency.

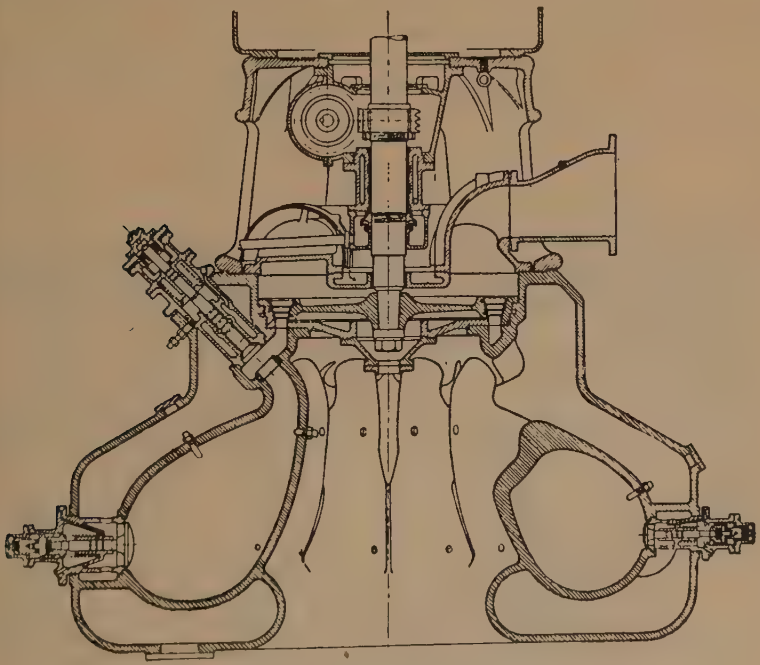

FIG. 11.—1,000-H.P. HOLZWARTH GAS TURBINE OF 1910

Barber and Parsons proposed machines working with continuous combustion at effectively constant pressure as opposed to the intermittent explosions of the internal-combustion piston engine. It is a little unexpected therefore to find that one of the earliest gas-turbine engines employed intermittent combustion of the piston-engine kind. This was the 50 h.p. Holzwarth[2, 3] gas turbine of 1908. Another Holzwarth engine, this time of a nominal 1,000 h.p. (Fig. 11), was completed in 1910 by Brown Boveri of Mannheim, where it ran experimentally before being moved to the Thyssen works at Mulheim. A 500 h.p. machine was built in 1920 for the German State Railways and, in 1925, a 5,000-kW plant was built for the Muldenstein power station. The German Brown Boveri Company have since built other Holzwarth gas turbines but, despite the theoretically high efficiencies possible, the type seems unlikely to survive. It is interesting to record, however, that Professor Belluzzo[4] reported as late as 1933 on very interesting results with small Holzwarth-type gas turbines, in which gas temperatures of 1,300°C were sustained with water-cooled turbine blades (see p. 149). A different kind of intermittent-combustion engine (see p. 142) employing a turbine—the free-piston engine—is undeniably doing well today, but the main stream of development and construction appears undoubtedly to be on the continuous combustion type, foreshadowed by Barber and Parsons and given a tremendous impetus by Whittle and the 1939–45 war.

The first gas turbine to work on the constant pressure cycle appears to be the 25 h.p. machine of Armengaud and Lemale, built at the Paris works of the Société des Turbomoteurs in 1904. This was a combination of a modified de Laval steam turbine, a combustion chamber burning atomized petroleum fuel, and a Rateau centrifugal compressor.[2] The same engineers built a 300 h.p. machine in 1906. In this the turbine was a two-row Curtis wheel and the twenty-five-stage compressor was of the Rateau centrifugal type built by Brown Boveri. The engine ran but was unable to provide effective shaft power.[2, 3]

Progress along what has proved to be the main stream of development was, for a quarter of a century after the work of Armengaud and Lemale, very tentative. Among the most practical contributions in this period, though in the field of auxiliaries and not of prime movers, was that of Buchi, the Swiss engineer who started in 1905 to work on the supercharging of the diesel engine by means of a blower driven by a turbine working in the exhaust stream of the diesel engine. This scheme has achieved widespread application and was applied to aircraft piston engines notably by Rateau and the American, Sanford Moss.

Developments contributory to gas-turbine progress came, however, also from the field of industrial compressors. In 1901, in his patent No. 3060 of that year, Sir Charles Parsons developed the axial-flow compressor ideas of the earlier patent I have mentioned. A compressor of this kind was built in that year (Fig. 12) and in general appearance was very like a gas turbine axial compressor of today. Other machines of the kind were built for industrial duty, but in those early years before the first world war the centrifugal type of compressor, successfully developed by Rateau, was more efficient.

The modern gas turbine, which now takes its place as the fourth of the great prime-mover systems, springs from a movement which is first clearly apparent in the middle 1930s. The chief influences at work were Swiss and British, though the work of George Jendrassik in Hungary, so little known at the time and, indeed, inadequately appreciated until quite recently, was contemporary and remarkable.[5]

The Swiss and British efforts were different in character and in sources. The Swiss developments had industrial objectives and were largely due to the Brown Boveri Company and its chief engineer, Dr Adolf Meyer. The British work was aeronautical in origin and its achievements were originally all of aeronautical objectives. In practical achievement the Swiss came first. In magnitude of achievement, the direct result of the British work has been stupendous, since it has produced a new era in flight.

GENESIS OF THE MODERN GAS TURBINE

In Britain

The story of the British effort has been told several times, but sagas are justifiably re-sung.

As I have pointed out elsewhere,[6] the work which led to the British aircraft gas-turbine engine came from two main streams and exhibited two very interestingly different methods of applied research. This difference, which I do not intend to labour in the Paper, can be described roughly as the contrast between the steady build-up of information by the study of the performance of components on the one hand, and the testing of the integrated conception of a genius on the other.

It was in 1926 that A. A. Griffith, whose extraordinary versatility has contributed so much to a wide variety of engineering sciences, produced, at the Royal Aircraft Establishment, an aerodynamic theory of turbine design based upon the flow past aerofoils as distinct from flow through passages. The consideration of this theory by the experts of the Air Ministry and the Aeronautical Research Committee led to his being authorized to do two interesting sets of experiments. One of these produced the first Royal Aircraft Establishment tests, reported on in 1928, on cascades of aerofoils representing compressor and turbine blading. The method of testing groups or cascades of stationary blades in specially designed small wind tunnels has since been developed and is widely used by gas turbine designers.

The second set of experiments was done with a tiny turbo-compressor (Fig. 13) which consisted essentially of a single-stage turbine and a single-stage compressor mounted on the same shaft and driven by an exhauster drawing air through the

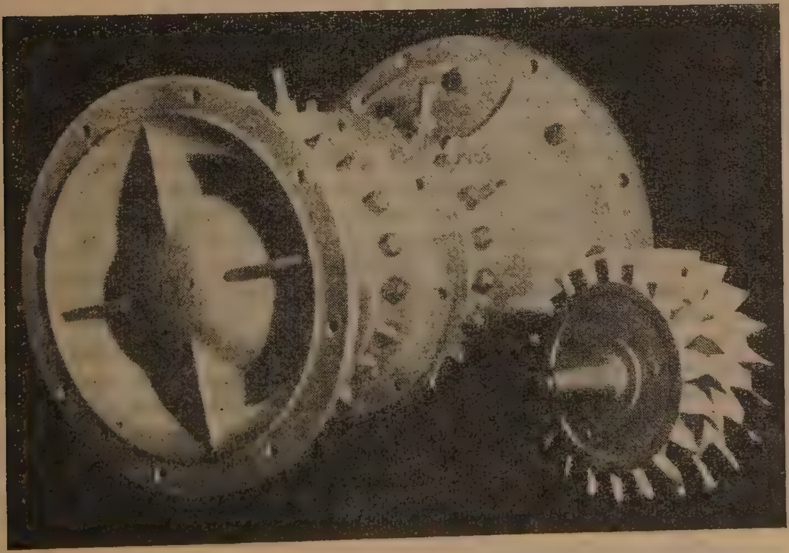

FIG. 13.—MODEL TURBO-COMPRESSOR TESTED AT THE ROYAL AIRCRAFT ESTABLISHMENT, 1929

casing. The overall efficiency of nearly 91% was remarkable, particularly as the blade heights were only about ½ in. and the overall diameter of the rotor barely 4 in.

In November 1929, Griffith put forward the merits of the gas turbine driving an airscrew as an aircraft power plant. He concluded that it would be lighter, more compact, and more efficient than the piston engines then projected. The particular scheme he suggested was based on contra-rotating wheels, which was a very advanced conception, and he proposed the construction of a rig to test it. The panel of experts which considered these ideas did not advocate the construction of an engine, but they recommended in April 1930 that the test rig be built for analysing flow, pressure, and temperature distribution. They considered also that combustion experiments should be done.

I have never understood why these recommendations were not put into effect. For some reason a sterile, almost blank, period ensued. For 6 years no work on axial compressors was done at the Royal Aircraft Establishment, and the only work in any way associated with gas turbines was the continuation of experiments done in the late 1920s on exhaust turbo-superchargers for piston engines.

In 1936, however, a second period of activity began. The Establishment began to build an axial-flow compressor, and, in association with the C. A. Parsons, Metropolitan-Vickers, Armstrong Siddeley, and General Electric Companies, a number of practical compressor experiments and design studies were made. One of the design studies arose out of the Metropolitan-Vickers collaboration and related to a low-pressure axial compressor driven by a low-pressure turbine in series with a high-pressure compressor and a high-pressure turbine, with a separate power turbine driving the airscrew. The high-pressure part (Fig. 14) of this was built. The compressor of it ran in December 1939, the turbine in May 1940, and the complete high-pressure unit in October of that year. The performance of this quite large axial compressor (18½-in. max. dia. over the blading) was excellent (its overall efficiency was approximately 87%).

The collaboration with C. A. Parsons resulted in the testing of several compressors, and the build-up of very valuable design information.

A small version of the contra-rotating turbo-compressor proposed by Griffith in 1929 was at last built by Armstrong Siddeley in 1939 to the aerodynamic design of the Establishment, where it was tested in 1940. This machine (Fig. 15) is still of very great interest. It consists of a number of co-axial wheels, adjacent wheels rotating in opposite directions. Each one carries an annulus of compressor blading, and an annulus of turbine blading. The air inhaled at one end is compressed in passing through the successive annuli of compressor blading, and is burned with fuel in a combustion chamber at the other end. From the combustion chamber the gas passes through the successive annuli of turbine blading and issues therefrom to provide a jet or to drive a power turbine.

The idea is extremely attractive in its compactness, its elimination of static blading, and opportunity for stress reduction. The 11-in.-dia. experimental machine tested in 1940 had an indifferent performance, but one cannot help regretting that it was not tried when it was first proposed. When it finally did run, however, tremendous happenings had occurred in the aircraft gas-turbine field, with more immediate prospects of early practical achievement, and regrettably, except for one interesting return to this kind of design in another experiment by Messrs Rolls-Royce, the design was not developed.

Of these tremendous happenings I shall say more shortly. They were connected with Whittle's success with his jet-propulsion engine. They had the effect of making

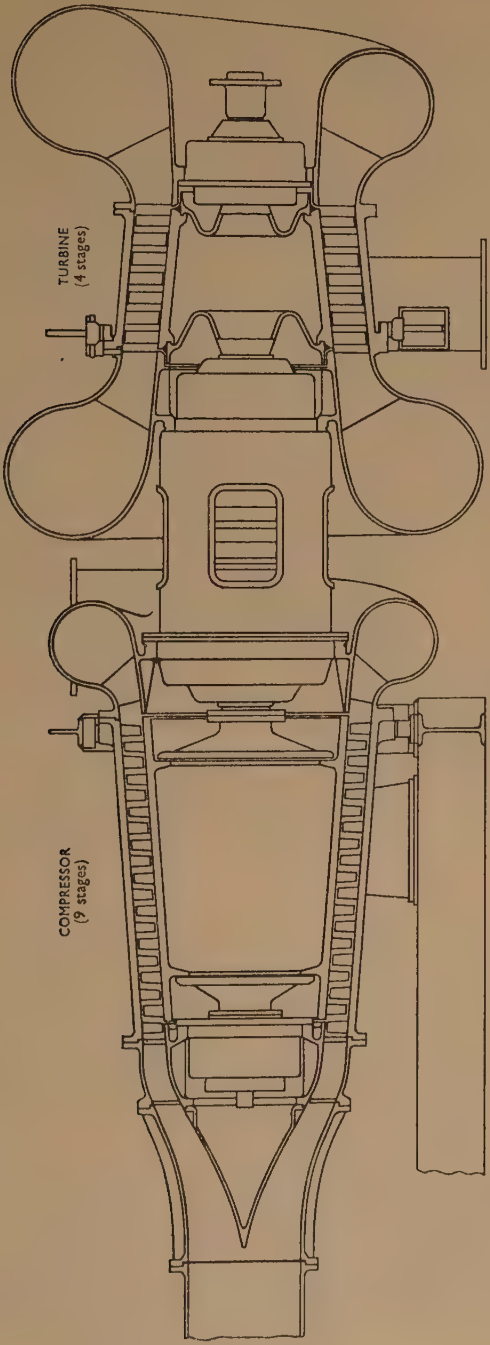

FIG. 14.—THE R.A.E.-METROPOLITAN-VICKERS B.10 TURBO-COMPRESSOR

those concerned at the Royal Aircraft Establishment with the development of the axial compressor gas turbine turn their ideas away from the driving of an airscrew to the provision of a jet. Consequently, the major gas-turbine project with which the R.A.E. was concerned in collaboration with Metropolitan-Vickers at the beginning of the 1939–45 war, a straightforward machine with a single compressor, an annular combustion chamber, compressor turbine, and power turbine, designed to give 2,000 s.h.p., began to slow down to its final abandonment, and was replaced by concentrated work on an axial-compressor jet-propulsion engine, which finally ran in December 1941. This, which became the Metropolitan-Vickers F.2 design, had a nine-stage axial flow compressor, an annular combustion chamber, and a two-stage turbine (Fig. 16). It developed finally into the Metropolitan-Vickers "Beryl."

Whilst Griffith and Constant—the latter whom I now mention for the first time, but who, beginning as Griffith's assistant and continuing today as the Director of the National Gas Turbine Establishment, has made one of the most important individual contributions to gas-turbine progress—were proceeding with limited resources and encouragement on their axial compressor development, another struggle against orthodoxy was in progress.

Frank Whittle, basing his proposals on thoughts developed from a 1928 thesis and his historic patent of January 1930 (Fig. 17), submitted to the Air Ministry in that year a plan to develop a jet-propulsion engine. Not only did Whittle's schemes differ from Griffith's in proposing jet reaction and not an airscrew as the means of propulsion, but in proposing a centrifugal and not an axial compressor. There was no practical outcome, however, nor was Whittle's approach to engineering firms more encouraging. The period of disappointment and frustration was the same period as that I have referred to earlier, when no work on axial compressors was done.

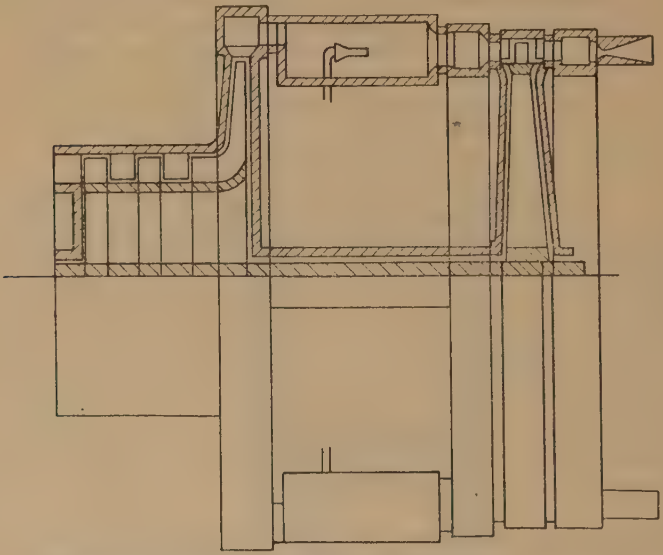

Fig 17.—Schematic diagram taken from Whittle's British Patent of January 1930

However, two friends of Whittle's succeeded in obtaining sufficient financial backing to form Power Jets Ltd in March 1936. This small company placed an order in June of that year with the British Thomson-Houston Company for most of the parts of a jet-propulsion gas turbine with a double-sided single-stage centrifugal compressor of pressure ratio 4:1, and a single-stage turbine. For the combustion system Whittle enlisted the aid of Laidlaw, Drew & Company.

The courageous policy of going straight to the building of a complete engine achieved the success it deserved. Tests on the engine began on the 12th April, 1937, and continued with intervals until the 23rd August. The success achieved impelled the Air Ministry to place contracts with Power Jets Ltd for a report on these tests and for further experiments. The engine was reconstructed, and in its new form ran on the 16th April, 1938. This series of tests was brought to a conclusion on the 6th May by a turbine blade failure. Further combustion testing and reconstruction, this time under Government contract, were put in hand, and the third version of the engine started tests in October 1938. This remarkable machine was used for experiments until February 1941, when it was wrecked by a turbine failure.

Long before this, however, in the summer of 1939, the Air Ministry had concluded that Whittle had in essence a practical aircraft engine. They therefore commissioned Power Jets Ltd to supply an engine for flight tests, and the Gloster Aircraft Company to design and construct the experimental aeroplane around it. This aeroplane (Fig. 18) was, and is, known as E.28/39, the number of its specification—the twenty-eighth (Experimental) Aircraft Specification of 1939. As a vehicle it has claims to fame equal to those of Stephenson's "Rocket" and Parsons' "Turbinia," and it is perhaps a pity that its sponsors did not choose a more attractive name for it.

The flight engine (Fig. 19) was designed by Whittle, with the help of the engineers of the British Thomson-Houston Company, who made it. It flew for 17 min on the 15th May, 1941, and continued to fly without mishap for a total time of 10 hours 28 min. The engine was then dismantled and found to be in excellent condition, having run in the air and on the ground a total of 39 hours 57 min. This historic engine had a thrust of 850 lb. at 1,650 r.p.m., its maximum speed.

Its installed weight was 623 lb. The all-up weight of the aeroplane at take-off with 81 gal of paraffin fuel was 3,691 lb. The designer of the aircraft was W. G. Carter. The pilot was the late W. E. G. Sayer.

Elsewhere

By the time the E.28/39 flew, the Swiss had made as much technical progress in the less spectacular direction of the industrial development of the gas turbine.

The earliest gas turbines of all were conceived as steps towards power-station machines generating electrical power, and the first practical gas turbines were of this kind. Probably the first major operational success of the stationary gas turbine, however, apart from its use as a diesel-engine supercharger, was as the pressurizing element in the Velox boiler.[3] The first Velox boiler, produced in 1930, employed a Holzwarth gas turbine, but in all subsequent examples of this compact and ingenious steam raiser, which runs on gaseous or liquid fuel, combustion takes place under pressure provided by an axial compressor driven by a multi-stage turbine operated by the products of combustion.

Experience gained by the Brown Boveri Company with Velox boilers, superchargers, and industrial turbo-compressors (see p. 147) led to the successful development of their prime movers.

The contribution of this great Company in the stationary-gas-turbine field, led technically in the pioneering days by Dr Adolf Meyer, dwarfs all others. There have been so many notable Brown Boveri gas turbines that it is possible to mention only a few. Probably the first gas turbine in the current tradition (a term which I use chiefly to exclude the Holzwarth machines) to go into power station service was the 4,000 kW stand-by set which the company installed at Neuchâtel in 1938. They supplied another set of 2,000 h.p., for experimental studies, to the Royal Aircraft Establishment just as the 1939–45 war was beginning. I well remember the installation of this machine and the experimental running achieved with it at Farnborough.

By this time too the Company were well on the way to producing the first gas-turbine railway locomotive, which began its trials in 1942.

All the Brown Boveri gas turbines were, and have since been, of the open-cycle kind. At the time of which I am speaking, however, the Escher Wyss company was beginning its notable development of the closed-cycle gas turbine (Fig. 4) in which the heating of the working fluid, usually air, which circulates and is not discharged to atmosphere, is done by "external" means as in the steam engine, in a boiler. External heating is of course possible also in the open cycle (Fig. 3). It has not yet been exploited in the gas turbine to the same extent as internal combustion, but its significance in the application of nuclear energy to the gas turbine is obvious.

The Sulzer company in Switzerland has concentrated on a cycle of operation which ingeniously combines both open and closed cycles. The Oerliken company has concentrated on the development of highly efficient centrifugal compressor gas turbines and on the study of combined steam and gas turbine installations. The operational outcome of the early work of the Escher Wyss, Sulzer, and Oerliken companies has, however, been in the post-war years.

To do justice to all the pioneers of the modern gas turbine is very difficult. The name of Professor Rateau will never be forgotten for its association with the development of centrifugal compressors. As I have remarked, he supplied one for the pioneer machine of Armengaud and Lemale. Subsequently, the Société Rateau did a great deal to advance the art of supercharging, of boilers as well as aircraft engines, by gas turbo-compressors, and their work in this field led in 1939 to plans for a 10,000 h.p. gas-turbine set for a torpedo boat. These plans were thwarted by the German occupation and though the set was made it was never fully tested. One other pioneer has been nearly lost to history. In the same year that Whittle began testing his first engine, George Jendrassik[5] began tests on a 100 h.p. gas turbine at the Ganz works in Budapest. It had a ten-stage axial compressor driven by a seven-stage turbine with a maximum speed of 16,500 r.p.m. A heat exchanger was employed and the pressure ratio was 2·2. In officially observed tests an overall thermal efficiency of 21·2% was obtained with a maximum temperature of 475°C at 98·5 h.p. Many hundreds of hours running were accomplished on this machine, some tests being run at 540°C.

Jendrassik designed two more gas turbines, both of which began tests in 1941. One was a 1,000 b.h.p. turbo-propeller aircraft engine, which, through combustion trouble, did not exceed 400 b.h.p. The project was abandoned before Jendrassik could try a better combustion system.

The other gas turbine, of 300 b.h.p., was intended for rail-car propulsion and other industrial duties. This machine employed a compact plate-type heat exchanger and a free-power turbine. The war prevented the development of this interesting engine also.

Like so many pioneers, Jendrassik was unfortunate. Had he lived farther west—or indeed, farther east—the outcome of his remarkable work might have been very different. As it is, his influence on the main streams of gas-turbine development was small, but the quality of his work shows that, in different circumstances, he would have given an immense stimulus to progress. Today, his designs and patents still deserve study.

From 1948 to his premature death in 1954 at the age of 55, Jendrassik worked in the United Kingdom. Even so, his early work did not get the notice it deserved and, regrettably, our appreciation has largely been too late.

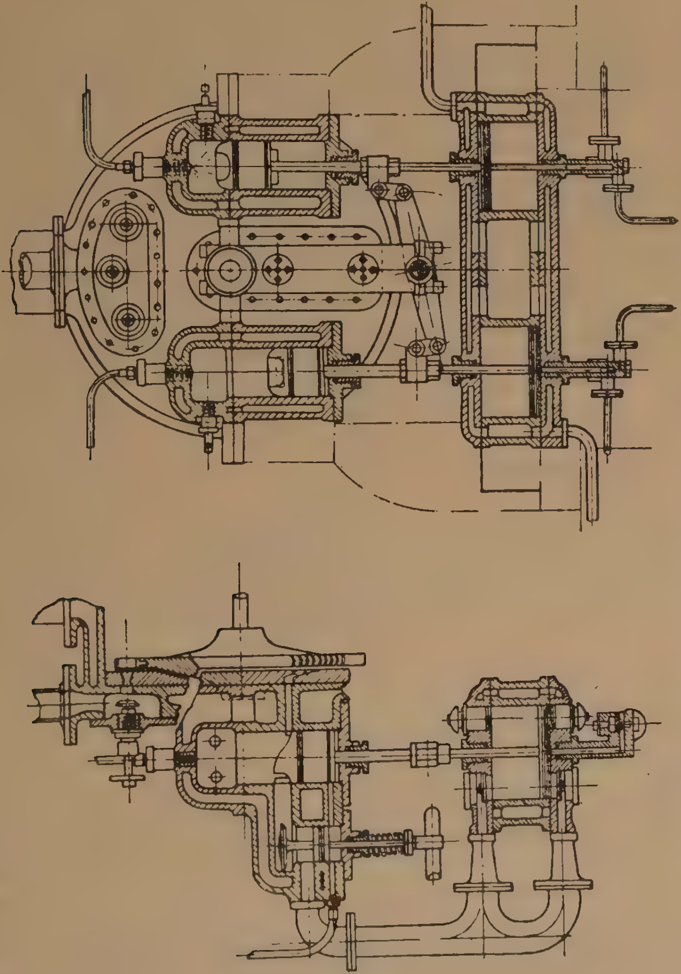

FIG. 20.—COMBINED OIL ENGINE AND GAS TURBINE OF BARRON, DALE, AND PACKHAM (BRITISH PATENT NO. 214,680 OF 1924)

There is one other development which should be mentioned at this stage, that of machines in which expansion is through a turbine but in which compression and combustion are achieved in cylinders with reciprocating pistons. I hesitate to allot priority in this field, but am unaware that C. B. M. Dale and his colleagues Barron and Packham, who began work in 1922, were anticipated. Their engine (Fig. 20) originally employed "free" pistons, but they were later connected by a crank system. Konig quotes[8] for this engine an efficiency of approximately 10%, achieved in the spring of 1924. Dale in a lecture[7] in 1944 refers to this work with great modesty and states that the venture perished through lack of funds. How many other examples of British ingenuity, inventiveness, and initiative have perished similarly? It is interesting to speculate how far Whittle would have got without the threat of war.

However, the *idea* of Barron, Dale, and Packham did not perish. The Swedish Gotaverken company have developed it using a diesel engine as the so-called gas generator. Pescara applied the horizontally opposed free-piston engine, effectively a diesel engine without a crankshaft, as the gas generator. In this form the free-piston gas turbine (Fig. 21) developed by Huber and his colleagues in France and later by Alan Muntz & Co. in England, is proving a very significant development.[9]

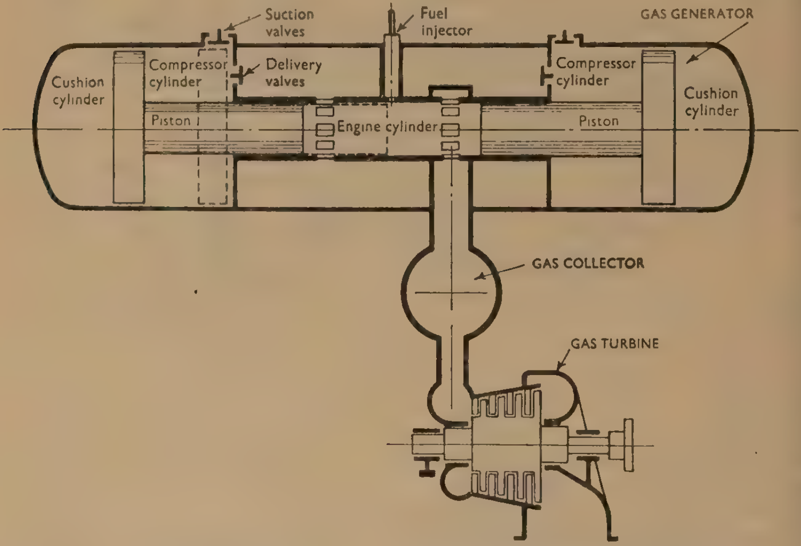

Fig. 21.—Diagrammatic sketch of a free-piston gas generator and gas turbine

THE GAS TURBINE IN AIRCRAFT

One day perhaps a history of the development of the gas turbine will be written. This is not it. This is the briefest of sketches. The history when it comes will be comparable in size with Gibbon's "Decline and Fall of the Roman Empire." I will dare to claim its theme no less noble. We must hope that its style will be equally distinguished. A tremendous proportion of it will inevitably be devoted to the aircraft application of the gas turbine because, undoubtedly, it is in the air that the gas turbine has achieved its most striking revolution. The reason is not far to seek.

The jet-propulsion version of the gas turbine has of its very nature produced a change in aircraft performance of such magnitude that a swing away from the piston engine was inevitable. Today it is roughly true to describe the air forces of the world as jet propelled, and civil aviation, again with the United Kingdom in the technical lead, is rapidly moving in the same direction.

I have already spoken of the beginnings of the aircraft gas turbine in the United Kingdom, and I have spoken of the first flight of a jet-propelled aircraft here. I wish I could have said that it was the first of such flights in the history of the world, but it was not. The Germans had probably begun aircraft gas-turbine work at much the same time as Whittle began in earnest (1936); that they knew of his ideas is certain, since the patents were not secret. It is not improbable that these patents stimulated the German engineer von Ohain, who designed an engine in which the compressor was a combined centrifugal and axial, in the late 1930s. A Heinkel aircraft employing this engine flew in May 1939. I feel we are justified in believing that if Whittle and the Royal Aircraft Establishment had not been frustrated by the tragic lack of interest in their gas-turbine work which I have mentioned, the Gloster-Whittle aeroplane would have flown before the Heinkel. The only consolation we have is that the Gloster-Whittle aeroplane was an immense success. The Heinkel was not. Nevertheless, although the first German gas turbine was not very successful, German jet-propelled aeroplanes were on operational service before British ones were. It must in fairness be said, however, that the German engines were put into production prematurely, and that their "life" in service would have been unacceptable in Britain.

In the United Kingdom the development continued from the centrifugal-compressor Whittle engine and the Metropolitan-Vickers axial engine. The brilliant early development which led to the first flight was followed by a period of considerable disappointment. In their early days the Whittle engines did not develop enough thrust for the second of our jet-propelled aircraft, the Gloster F9/40, to take off on the runways available to us here. The American Bell XP-59A aircraft, powered with similar engines made by the General Electric Company of Schenectady, operating from the almost infinite airfield of Muroc, was in fact the second jet-propelled machine on the Allied side to fly (October 1, 1942). The G.E.C. had built their engines from information and drawings which we handed to them, a collaboration which in fact anticipated the American entry into the war.

In England the Whittle W2B engine was developed not only by Power Jets but by the Rover Company and, ultimately, by Rolls-Royce, in whose hands it became known as the Welland, the operational engine of the Meteor aircraft which developed from the experimental F9/40. It was the first of the operational Rolls-Royce gas-turbine engines, all of which have been distinguished by the names of British rivers. I have elsewhere[6] given in some detail the technically glorious but rather complicated history of aircraft gas-turbine development during the war years. The Meteor with the Welland engines was the only Allied machine to become operational before the war ended, but most distinguished work was done by the Metropolitan-Vickers Company with their development of the Beryl engine, by the de Havilland Company with their development of Major Halford's H.1 (later the Goblin) engine, and by Armstrong Siddeley with their ASX. As the war ended the Bristol Company's Theseus was coming to fruition.

On the German side we had to contend with the Heinkel powered with Junkers 004 engines, and the Messerschmitt powered with BMW 003 engines. These were both plain axial-compressor engines, the type which in military aviation has prevailed.

It is a little ironical that the brilliant work of Whittle which gave the whole impetus to our wartime development, and seemingly that of the enemy as well, was concentrated on engines with centrifugal compressors (Fig. 22), which will probably never appear again in high-performance military aircraft. In the civil field they still possibly have advantages, and certainly the most successful civil gas-turbine engine operating at the present time is the Rolls-Royce Dart (Fig. 23) in which compression is achieved by two centrifugal compressors in series. This, I should emphasize, is not a jet-propulsion or turbo-jet engine. It is a gas turbine delivering shaft power to an airscrew—a "turbo-prop" engine—as conceived by Griffith before Whittle's jet propulsion proposals. Turbo-prop engines are particularly suited to the economics of civil aircraft required to fly in the speed range 300–450 m.p.h.

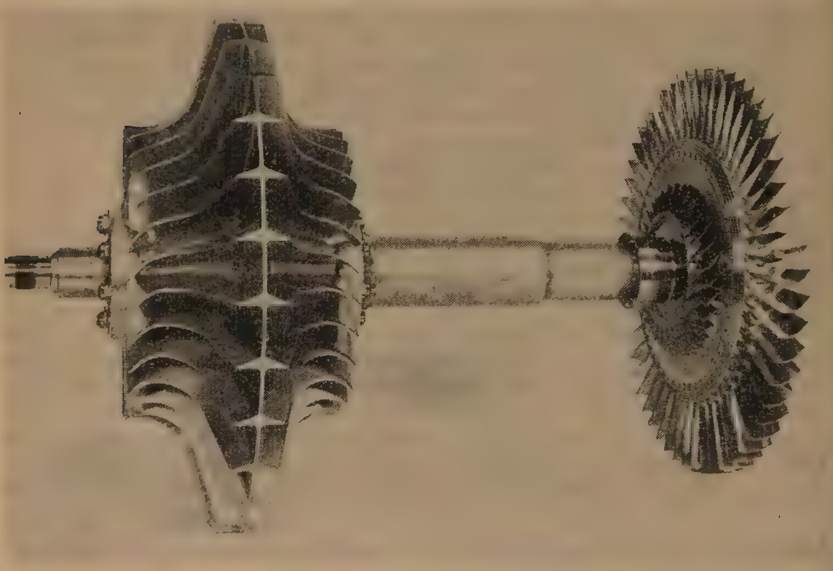

FIG. 22.—CENTRIFUGAL IMPELLER AND TURBINE WHEEL OF THE WHITTLE W2B ENGINE

In the great history which I have mentioned, and which I look forward to reading very critically in my declining years, there will appear a galaxy of achievements which I can refer to here only by lists of names. In America the work which began in the General Electric Company under the guidance of Standerwick and Warner was followed by engines at Allis-Chalmers, Westinghouse, Allison, Pratt & Whitney, and Wright. In Germany there were Junkers and BMW; in France the Cie Électro-Mécanique, SNECMA and Turbomeca; in Sweden Flygmotor and STAL; in Canada A. V. Roe & Co., successors to the wartime Turbo-Research Ltd. There has been an immense proliferation and variety of engines since the war ended, and any attempt to give a complete picture this evening would fail. What is immensely gratifying to record is that the aircraft engine industry of the United Kingdom, profiting from the technical lead which Whittle gave it, and, while competing, co-operating through the Gas Turbine Collaboration Committee (see Appendix III), has kept ahead of the rest of the world in the development of the aircraft gas turbine. British firms

Fig. 31.—M.G.B. 2009

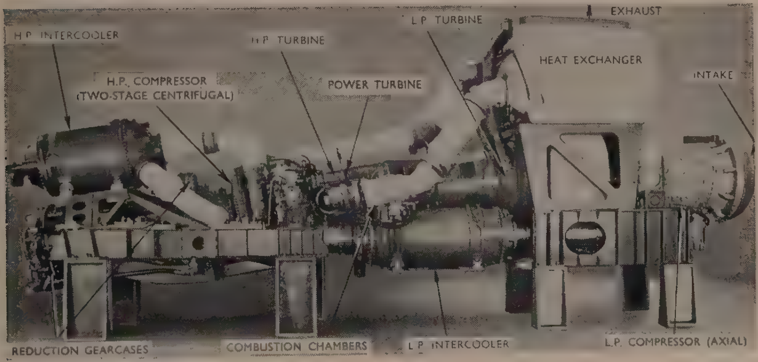

Fig. 32.—Rolls-Royce R.M.60 engine for H.M.S. *Grey Goose*

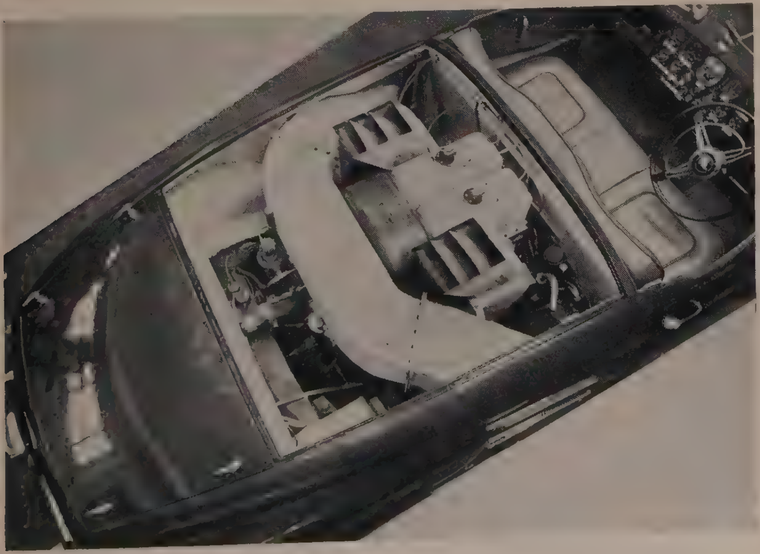

Fig. 34.—Installation of the Rover automobile gas turbine

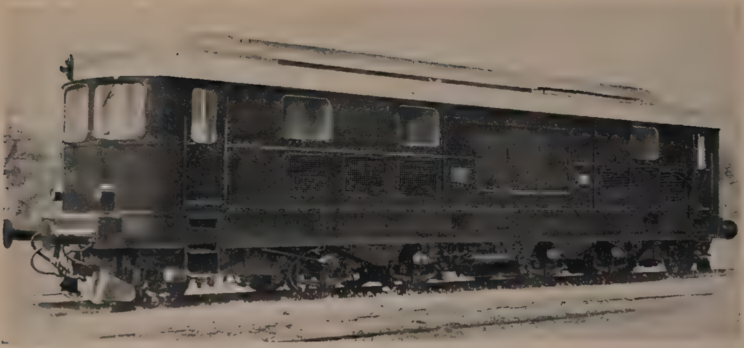

Fig. 35.—The first Brown Boveri gas-turbine locomotive (1942)

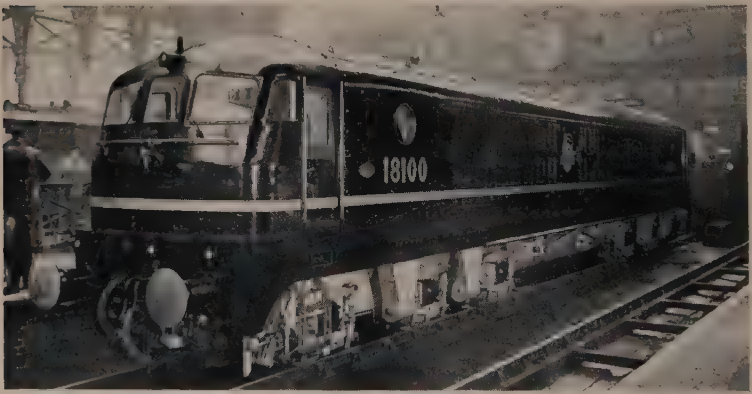

Fig. 36.—The British Railways 3,000-h.p. Metropolitan-Vickers gas-turbine locomotive

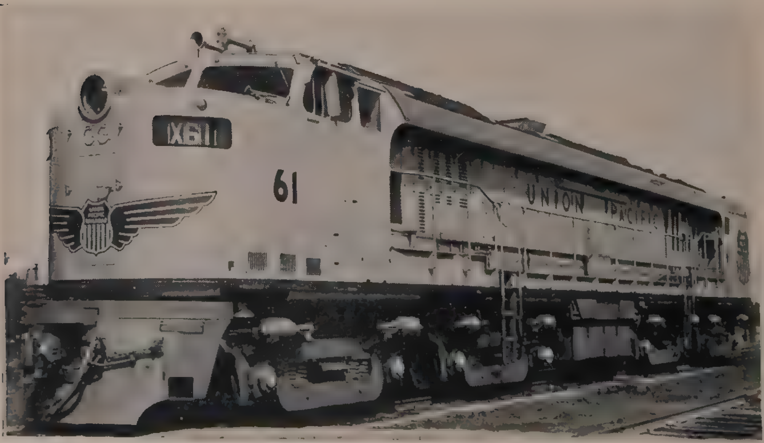

Fig. 37.—The Union Pacific Railroad's General Electric gas-turbine locomotive

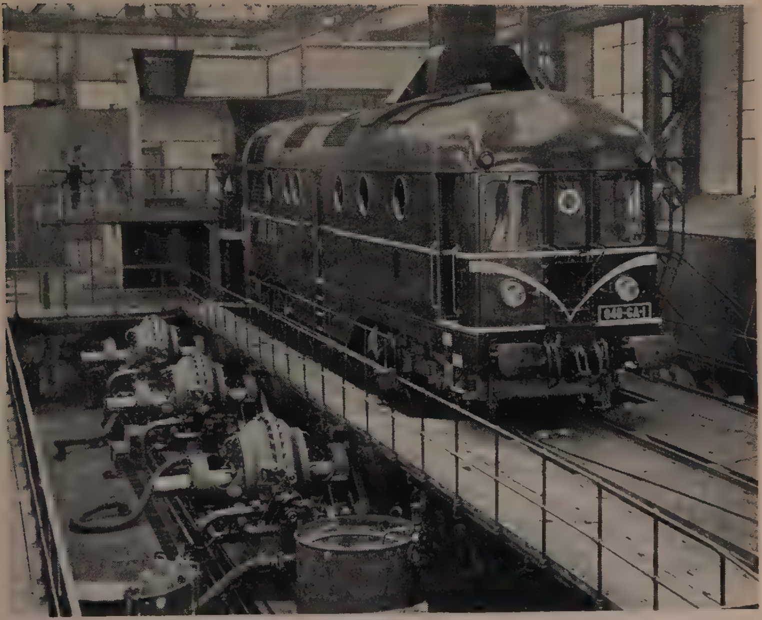

Fig. 38.—The Renault-SIGMA free-piston gas-turbine locomotive of the S.N.C.F. on the test bed

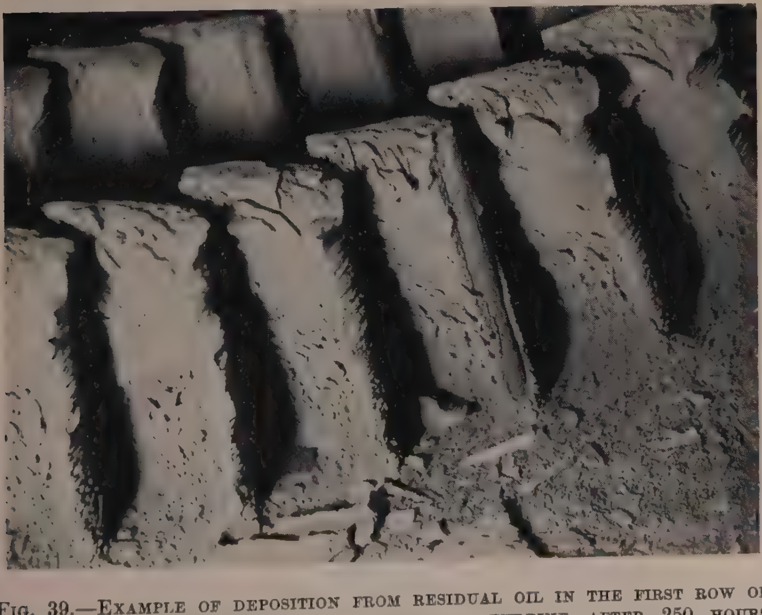

Fig. 39.—Example of deposition from residual oil in the first row of stator blades of a low-pressure turbine after 250 hours without additives

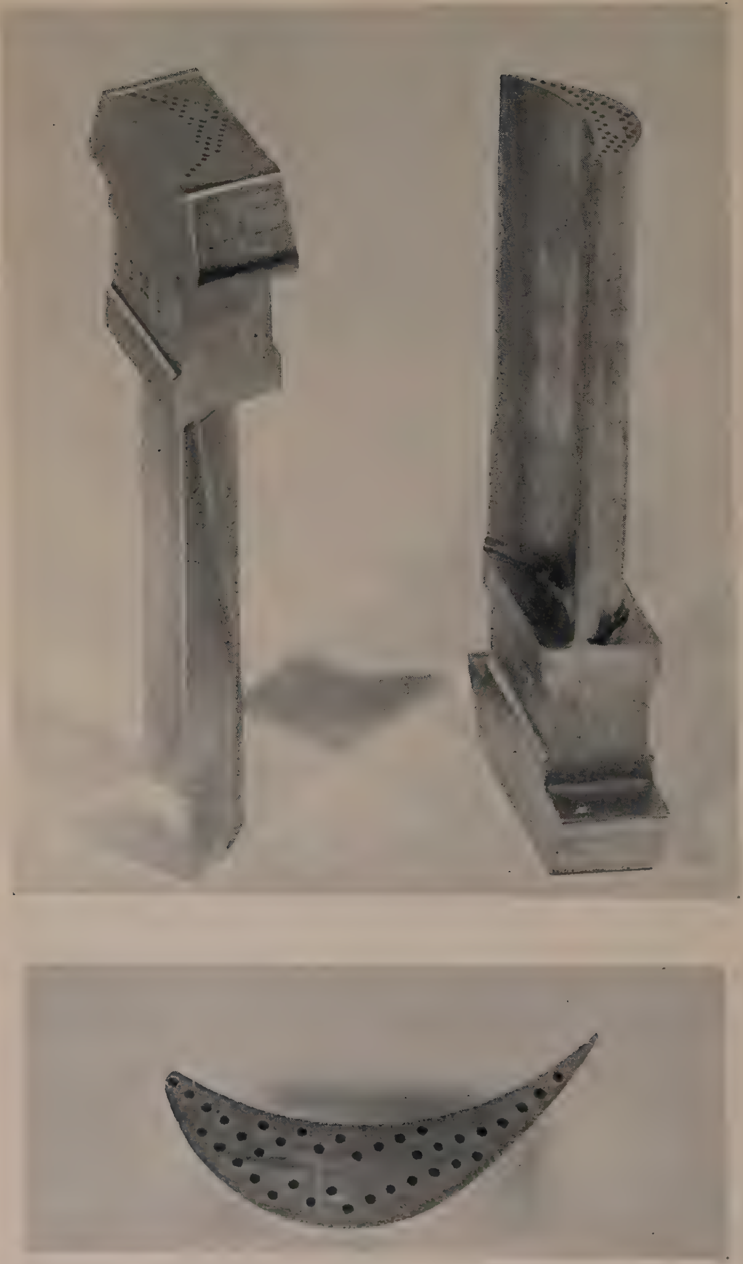

Fig. 40.—Turbine blades with cooling passages

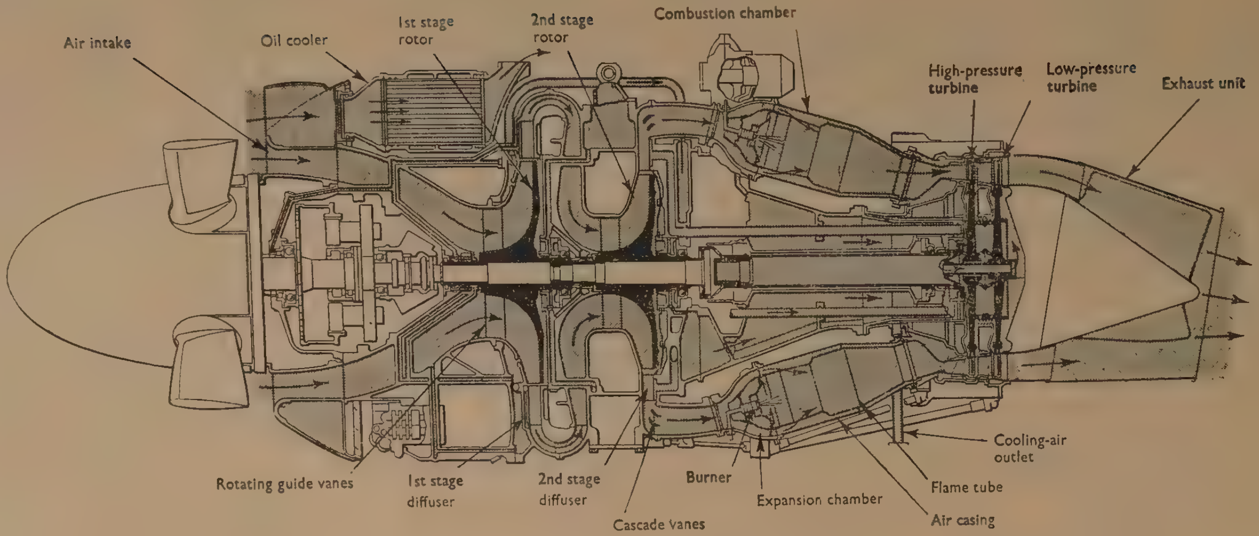

FIG. 23.—GAS-FLOW AND COOLING-AIR DIAGRAM OF THE ROLLS-ROYCE "DART" ENGINE

have sold licences for their aircraft gas turbines to the United States, to Sweden, to Denmark, to Switzerland, and to France. There are other good aircraft gas turbines besides the British ones, but there are none better. And as I am lecturing in England, in honour of an Englishman, it is not inappropriate to illustrate the present state of the art by reference to British engines.

Of the straightforward jet-propulsion engines those which are chosen to engine the three great aeroplanes of our new bomber fleet, the Vickers Valiant, the Avro Vulcan, and the Handley Page Victor, are the Rolls-Royce Avon, the Bristol Olympus, and the Armstrong Siddeley Sapphire. The Sapphire (Fig. 24) is in fact a descendant of the Metropolitan-Vickers Beryl, and its early development was in the hands of that company. It has been brought to its present high pitch of excellence by Armstrong Siddeley. This engine and the Rolls-Royce Avon (Fig. 25) are both single-shaft engines (Fig. 5), but the Olympus (Fig. 26) is a two-shaft engine. In other words it has two rotors, which run at different speeds (Fig. 27).

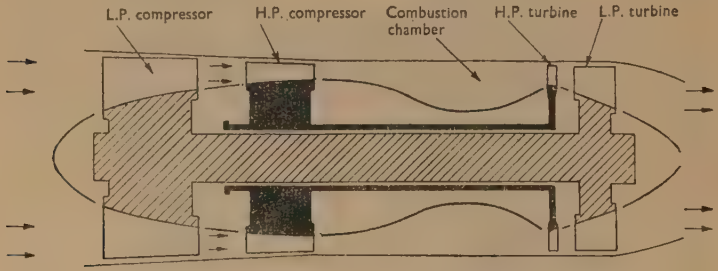

FIG. 27.—JET-PROPULSION ENGINE WITH TWO AXIAL COMPRESSORS

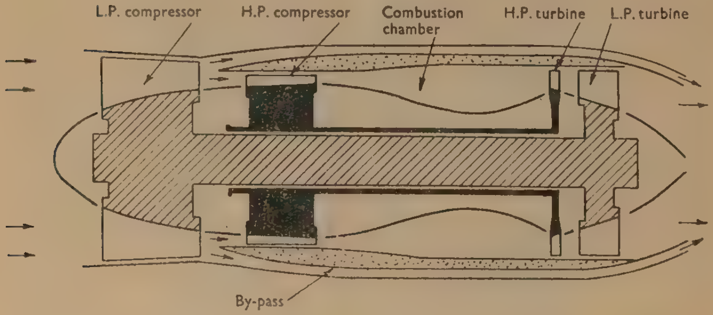

FIG. 28.—BY-PASS JET-PROPULSION ENGINE

Britain's most successful civil aeroplane at the present time is undoubtedly the Vickers Viscount. It employs the engine to which I have already referred, the Rolls-Royce Dart. This 1,500 h.p. engine has an overhaul life of 1,000 hours, a record not bettered by any other aircraft engine of any kind. It is at the basis of the Viscount success. I wish I had the time to discuss the many excellencies of the huge 15,000 lb.-thrust de Havilland Gyron, or the Napier Eland, on either of which I could interest you for an hour, but I will instead refer to the Rolls-Royce Conway because it is different from any other engine at present in development. It is a by-pass engine of 13,000 lb. thrust. A by-pass engine is so arranged (Fig. 28) that some of the air inhaled at the front is not burned but is ejected at the rear with the products of combustion. The effect of this is that for a given expenditure of fuel a greater change of momentum is produced and the mean jet velocity is reduced. This means not only improved economy but a diminution in noise level, so that engines of this type appear to be specially appropriate in civil aviation. We shall have to wait a while for the first airliner to employ them however. Meanwhile, we hope that the great achievements of the Viscount will be emulated by the Bristol Britannia with its Bristol Proteus engines, by the new de Havilland Comet with its Rolls-Royce Avon engines, and the recently announced Vickers Vanguard. The Vanguard will have four Rolls-Royce Tyne turbo-prop engines, each of about 4,500 equivalent horse-power. This is another two-shaft gas turbine, with the highest pressure ratio yet announced in the United Kingdom for an aircraft engine, 12:1.

THE GAS TURBINE IN POWER STATIONS

The early work of the Brown Boveri Company to which I have referred quickly led to serious commercial developments. The Company has supplied gas turbines of various sizes, for electric power generation, several in the 10,000–13,000 kW range, to many countries, but probably their most notable installation is that in their own country at Beznau (Fig. 29). Here is a complete gas-turbine power station of 40,000 kW, and more than 30% thermal efficiency. There are two engines, one of 13,000 kW and one of 27,000 kW. The former, commissioned in January 1948, has a pressure ratio of 8·8:1. The latter, commissioned a year later, has a pressure ratio of 8·1:1. Whilst they are not base load plants, they frequently run with a high load factor in the course of their duty of supplementing the Swiss hydro-electric system. In the 5 months (about 3,600 hours) ending on the 31st March, 1954, they ran respectively 1,950 hours and 2,250 hours.[10]

The Escher Wyss Company supplied a 12,500-kW closed-cycle set to the St. Denis station of Electricité de France near Paris, and a set of similar power and character has been supplied by John Brown & Co., Ltd, of Clydebank to the Dundee power station.

The largest gas-turbine power station plants in the United Kingdom at present are the 15,000-kW Metropolitan-Vickers open-cycle set (Fig. 30) at the Stretford power station near Manchester and the 15,000-kW C. A. Parsons open-cycle set at the Dunston power station near Newcastle. The first of these first ran on load in August 1952 and the second a few weeks ago.

In all these sets, good thermal efficiency is obtained by the use of heat exchangers (Fig. 2) in combination with good component efficiencies. A thermal efficiency of 24·5% has, however, been achieved without a heat exchanger in the 9,000-kW set produced in 1955 by the Swedish STAL company by employing a pressure ratio of 10:1, unusual in an industrial set.

Small gas-turbine sets are numerous. In the United Kingdom a 2,500-kW set has recently been installed by Metropolitan-Vickers at Ashford for the Metropolitan Water Board for pumping duties, and others of similar size are under construction by other companies. Ruston & Hornsby have made several 1,000 h.p. sets which, with appropriately different combustion chambers, have run on distillate oil, residual oil, natural gas, and peat.

A very interesting development in France has been the application to power-station duty of the free-piston gas turbine. A 1,250 kW unit with two free-piston gas generators and a single Rateau turbine has been installed at Rheims. It has run for more than 5,500 hours. Quite large powers are achieved by using several cylinders and there has recently been completed at Cherbourg for Electricité de France a 6,000-kW set employing eight gas generators.[9]

Electricité de France have supported also the deeply interesting experiments of Ernest Mercier * who has developed a special form of free-piston gas turbine which works in combination with a high-pressure steam generating system.

As a power-station unit, the ordinary gas turbine has a number of advantages, actual and potential. It can quickly be brought on to load, which makes it of particular value as a stand-by or peak-load set. As an open cycle machine, it demands no cooling water, so there is little restriction in siting it; it can be used in deserts and mountains where water is difficult to come by. Having no boiler, it should be more compact than a steam turbine. This advantage is to some extent offset by the need in many cases for a heat exchanger, but still, on the cost of housing, the gas turbine should win. The gas turbine's lower weight and size suggest a lower capital cost per kW, but its demand for high-quality materials and the relatively early stage of its commercial development have so far prevented any marked advantage of this kind yet being demonstrated.

The other great advantage, which must admittedly still be regarded as partially potential, is the ability of the gas turbine to run on almost any fuel. The majority at present run on oil, generally a residual oil. In the United States, however, gas turbines are used for pumping the great supplies of natural gas from Texas to the north; these run on the natural gas itself.

The need to use the indigenous fuel resources of the United Kingdom economically has led in recent years to an elaborate exploration under the general auspices of the government of the possibilities of running gas turbines on solid fuel and its derivatives.[11, 12] A 500 h.p. closed-cycle gas turbine has run for 1,000 hours on peat, for 500 hours on coal, and for 500 hours on a coal slurry. These achievements were by John Brown & Co. of Clydebank. An open-cycle English Electric set of 2,000-kW capacity has run on coal with a special form of combustion chamber from which most of the ash in the coal is withdrawn as liquid slag, in contrast to the more usual methods of momentum separation of dry ash. The combination of an open gas-turbine cycle with a very high pressure gas producer cycle is also being developed; this scheme permits a different and perhaps still more convenient method of gas cleaning.

More unexpected in character than any of these, however, is the English Electric gas turbine being installed at Stafford Colliery by the National Coal Board which can use the methane content (about 0·5%) of the ventilating air from the mine, and the Ruston & Hornsby gas turbine which runs on Scottish peat. The temptation to digress on to the story of the dewatering of peat to make it a suitable fuel for an internal-combustion engine is a great one which I am resisting, saying only that the success of an open-cycle gas turbine using this fuel depends on an economic system for removing about two-thirds of the water of this very damp substance, and such a system has been developed. I hope a lecture on it will shortly be read in London by the engineer chiefly responsible.

With all these advantages of compactness, independence of water, and omnivorousness it is perhaps a little surprising that, other than in combination with the steam

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* Mercier, a man of great gifts, died in July 1955. It is to be hoped that his colleagues will continue his work.

turbine,[26] the gas turbine has not developed more as a base-load plant. As a plant to be used for stand-by purposes and in unusual topographical conditions it has made its mark, but as a base-load plant it has still to do so. There is little doubt that Beznau could run as a base-load plant so long as some blade cleaning could be done at low-load periods. But even that station is small by modern power-station standards and no gas turbine competitive in power output with the current 100 mW and the coming 200 mW steam-turbine plants has been projected. Gas turbines of such size are certainly possible, but considerable development would be needed and a considerable investment would be involved.

The existing gas-turbine power station sets will all provide very valuable operational and technical information, but even the greatest gas turbineers amongst us would hesitate to regard them as precursors of huge sets of the same character in the United Kingdom. There have long been reasons for this, but there is now the additional reason that the mighty atom is in our midst and has begun to affect power station design. At present it produces low temperatures which can be used in low-performance steam turbines, but we shall quite undoubtedly find means of safely creating and safely delivering from the pile to the rotating machinery much higher temperatures, and when this has been done the rotating machinery will be a gas turbine. In fact one foresees in the atomic power station the apotheosis of the gas turbine.

THE GAS TURBINE IN SHIPS

The early work on the aircraft gas turbine was watched closely by the Admiralty and the development of the Metropolitan-Vickers F.2 engine had not gone very far before they decided that it would be a very interesting plant to adapt to the duties of the motor gunboat.[13] This type of craft needs to be able to cruise for long periods at relatively low speeds and to achieve very high speeds for short periods. This specification was met in MGB 2009 by retaining two of the standard high-economy petrol engines used in MGBs for ordinary cruising and installing the high-performance F2 gas-turbine engine, with a special power turbine and gear box, for high-speed running. In this form, known as the GATRIC, the engine developed 2,500 s.h.p. This vessel (Fig. 31) went to sea in August 1947, so, almost 50 years after Parsons had made history with *Turbinia*, MGB 2009 made history too; it was the first vessel to be propelled at sea by a gas turbine.

Trials continued for a period of 4 years and brought to light few troubles. Clearly the characteristics of the high-performance gas turbine were suitable for high-speed naval craft. Consequently the Admiralty ordered further plants from Metropolitan-Vickers based upon the "Beryl" aircraft engine, into which the experimental F.2 had developed. These plants, known as "G2," were installed in two fast patrol boats, *Bold Pioneer* and *Bold Pathfinder*, trials on which began in 1952. Each boat had two gas turbines, each of 4,500 h.p. Still more powerful plants of the same kind are in hand.

These gas turbines, however, are by way of being boost plants to give very high performance when required. They are not main propulsion machinery. The first gas-turbine plant capable of economical low-power cruising and high-speed running as well when required was the Rolls-Royce RM60 (Fig. 32) designed for H.M.S. *Grey Goose*.

This, to cover the performance range required, is necessarily a relatively complicated plant of 18·5:1 pressure ratio incorporating a heat exchanger and intercooling.

Two of these engines, each of 5,400 s.h.p., have been installed in the gunboat *Grey Goose* and are undergoing sea trials. Sir Charles Parsons would have been interested to know that the gas turbine installation gives 50% more power for half the weight and occupies 25% less space than the steam machinery it replaces.

The pioneer work of the Royal Navy in this field has not of course passed unnoticed abroad and other navies have begun to take a practical interest.

Generally speaking, naval propulsion machinery has to run for long periods at

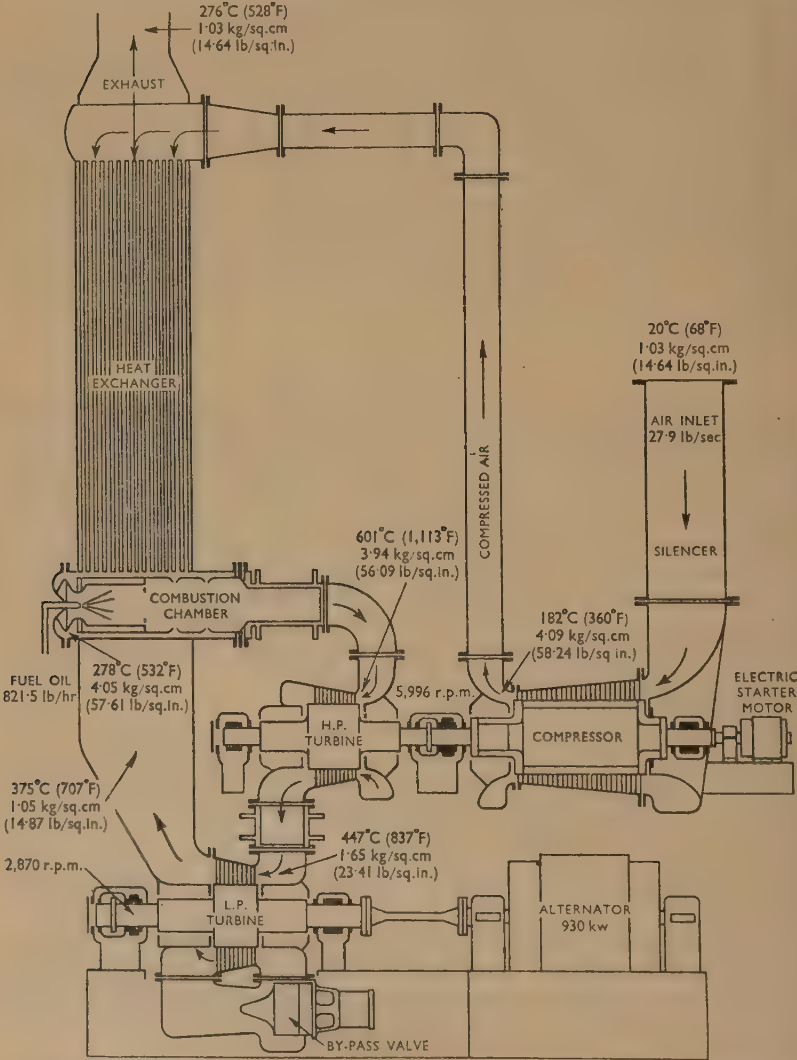

Fig. 33.—Diagrammatic lay-out of 1,200-h.p. gas-turbine set for the tanker *Auris*

low loads and short periods at high loads. In the mercantile marine the situation is quite different. There, long life at cruising speed is the demand. Consequently, the design of naval and mercantile gas turbines is likely to be very different.

In the mercantile field, as in the naval, the pioneer work has been British.[14] The very great progress made has been mainly due to the ability, drive, and enthusiasm of Mr John Lamb and his colleagues in the Shell Marine Research and Development Department, to the directors of the Shell group whose 12,000-ton tanker *Auris* has been the vehicle of experiment, and to Mr Forsling and the other engineers of the British Thomson-Houston Company who have been responsible for the gas-turbine installations in it.[15]

In 1951 the *Auris* became the first merchant ship in the world to be powered by a gas turbine (Fig. 33). The 1,200 h.p. gas turbine was not the only propulsion machinery aboard however. There were three diesel engines each of similar power. Nevertheless, the stage was set for extensive sea trials and the gas turbine has done 15,000 hours running. At least once, the South Atlantic was crossed under gas-turbine power alone.

This running has brought to light few problems, and it has provided opportunity for trials with residual oil fuel of high viscosity. To use such fuel without producing deleterious deposits in the turbine blading is a perennial problem to gas-turbine designers, but in the case of the *Auris* machinery some success in preventing deposition has followed the use of a finely divided powder, aluminium silicate, in the fuel. Altogether, the experimental running has induced a feeling of such confidence that the existing gas-turbine and diesel engines are to be removed and replaced by a single gas-turbine plant of 5,500 h.p. The gas turbine will be made by the British Thomson-Houston Co. Ltd and the reduction gearing by Cammell Laird & Co.

The new machinery will provide the power not only for propulsion, but for auxiliary and pumping services both at sea and in port. Commercial grades of boiler fuel will be used.

The French have applied the free-piston gas-turbine system successfully to marine craft, concentrating particularly on minesweepers and coasters.[16] The first of the coasters, *Cantenac*, of 850 tons, has been in service since 1953.

THE GAS TURBINE IN LAND VEHICLES

The first public appearance of a gas-turbine-driven road vehicle was in March 1950, when the Rover Company began a series of test runs with their car JET 1 at Silverstone. This car employed a Rover open-cycle gas turbine of about 110 b.h.p. Despite its number-plate it was not, of course, jet propelled. As much energy as possible was taken from the gas by the power turbine, which drove the back wheels of the car through a gear box giving about 16:1 reduction. The air was taken in at the sides of the car and exhausted just behind the seats (Fig. 34).

In the same year the Boeing 175 h.p. gas turbine was applied in the U.S.A. to a Kenworth truck of about 68,000-lb. loaded weight.

These first experiments emphasized what had been known of course all along by the designers, that for road work the chief problem was the achievement of fuel economy. This means in practice the development of suitable heat exchangers or high-compression engines. Despite the work of the Rover Company and the Austin Motor Company—whose gas-turbine car was first demonstrated in 1953—in England, the Boeing and Chrysler companies in the U.S.A., Laffly and Gregoire in France, and Fiat in Italy it cannot be denied that there is still some way to go before an attractive gas-turbine car appears. For the gas turbine to compete in the field of the private car, cheapness in first cost and upkeep, dependability, and economy are essential. Cheapness in upkeep is a gas-turbine characteristic and it is a reliable machine; cheapness in first cost is largely a matter of large production numbers; so economy in running is left as the major problem.

It may be that the gas-turbine car will never show the same figures for fuel consumption as the petrol engine, but it will be able to use cheaper fuel—possibly even residual fuel. In places where such fuel is cheap this may be a great advantage, particularly for commercial vehicles. In the commercial-vehicle field also, the possibilities of the free-piston gas generator combined with a gas turbine are interesting.

In the field of road traction, the future of the gas turbine is less clearly seen than in other fields, first because it is not yet clear what degree of advantage the gas turbine can offer and second because the competition from the petrol and diesel engines, which have reached so high a pitch of excellence, is so keen. That it will have a future, however, even though it may be specialized in character, is not in much doubt.

On the railways, too, the true value of the gas turbine cannot be quickly assessed. Competition with steam is not a factor. Steam is on the way out even in the United Kingdom, locomotionally the most conservative of all. The gas turbine is up against two well-developed and efficient rivals—electrification on high-density lines and diesel engines on the rest. The advantages of electrification for suburban and high-density main-line traffic seem to me so overwhelming that I feel the true problem is to decide upon the right kind of engine for secondary routes and low-density long-distance traffic.

Technically, the gas turbine can do the job. This was demonstrated by the Swiss during the 1939–45 war; they began to run the 2,200 h.p. Brown Boveri locomotive (Fig. 35) on their railways in 1942. A similar locomotive, and a Metropolitan-Vickers one of 3,000 h.p. (Fig. 36), have been running for some years on British Railways. But they are in competition with the conventional diesel engine, which uses about half as much fuel. On maintenance the gas turbine is superior and the decision which to use is greatly dependent on fuel cost. Where this is a less important factor than in England, as in the U.S.A., the gas-turbine locomotive has made an encouraging start. There are upwards of thirty of these built or building for the Union Pacific Railroad using General Electric–Alco turbines (Fig. 37).

The Renault Company in France believe in the free-piston gas turbine.[18] This has many of the advantages of the turbine with approximately the fuel economy of the conventional diesel engine. An interesting engine of this kind of 1,000 h.p. (Fig. 38) has been running since 1953 on the S.N.C.F. and has covered about 100,000 km; two 2,000 h.p. locomotives of the same kind are being built.

On the railway, the gas turbine offers high availability, since it is easy to maintain, has a capital cost comparable with the diesel, and gives good economy with a cheap fuel. There is, therefore, some case for a gas-turbine locomotive in the United Kingdom to run on coal. Its economy on this fuel would be much superior to that of the ordinary coal-burning steam engine. A gas-turbine locomotive of this kind—the first of its kind—will probably be running in the United Kingdom in a few months time. Its chassis is being built by the North British Locomotive Company and its prime mover, by C. A. Parsons, works on the open cycle but with external combustion [11] (Fig. 3). This unusual choice was made to ensure that the ash in the products of combustion does not go through the turbine. This, however, is not likely to be the final method of dealing with the coal-ash problem.

A locomotive is not often called upon to give its full power. It spends the greater part of its running life at less than half-load. The efficiency of the gas turbine at such low part loads is much reduced. This militates against high performance in the cases of all the locomotives I have mentioned. A way out of the difficulty would be to have more than one gas turbine unit; probably three or four per locomotive would be ideal. The first step in this direction has been taken by the Boeing Company in America; they have built a 30-ton locomotive with two small engines of 150 h.p. each.

THE GAS TURBINE AS AN AUXILIARY AND IN PROCESS WORK

The earliest practical application of the gas turbine was in the field of supercharging of piston engines. The exhaust from such an engine drives a turbine which in turn drives a blower which compresses the air delivered into the cylinders. Dr Alfred Buchi was first in the field, supercharging four-stroke diesel engines in this way in Switzerland in 1905. Rateau applied turbo-supercharging to aeroplane engines in 1916 and his work was taken up in the U.S.A. by Moss. Development of the system continues today on diesel engines and in the last of the aircraft piston engines.

Another famous auxiliary application of the gas turbine is Brown Boveri's Velox boiler, which I have already mentioned. These boilers have been installed throughout the world, not only in power stations, but also in French warships.

The earliest application of the gas turbine in chemical process work was the Brown Boveri turbo-compressor for the Houdry petroleum cracking process.[3] Supercharging from an external power source would have been economically unsound, but the process exhaust gases which would otherwise be wasted are expanded in a gas-turbine rotor which not only drives the supercharging compressor, but supplies electrical energy as well. The first application of the gas turbine in this way was in 1936 and it has been repeated nearly thirty times since.

The gas turbine generally has the property of being able to run on low calorific value fuel, and it is in consequence attractive as a means of using the energy in blast furnace gas to supercharge or "blow" the blast furnace. In this field too Brown Boveri have been pioneers. Their installation at Baracaldo in Spain is understood to have operated satisfactorily since 1952.[19] Another Brown Boveri plant running on blast furnace gas is the 7,250-h.p. set at the Arbed steelworks in Luxembourg.[19] Despite its specialized duty, this might be more correctly classed as a power station however, since, besides supplying compressed air, it directly generates electricity.

Since the 1939–45 war, gas turbines have been adapted to a wide variety of auxiliary duties. In small sizes they are having some success as drives for fire pumps and as auxiliary naval machinery. In this last class, a set as large as 1,000-kW has been supplied to the Admiralty, but it is probably most correctly regarded as a small sea-going power station.

GAS TURBINE RESEARCH

Probably it will be agreed that, although not comprehensive, my lecture so far has illustrated the wide variety of applications of the gas turbine. I have said little, however, about the research and development work behind the achievements I have mentioned, and nothing about the research and development work now being done on which future gas turbines will be based. It is not easy to do justice to this effort, spread as it is over many countries, some of it very fundamental in character, some of it of the nature of *ad hoc* development, and most of it somewhere in between. I believe a picture—a cartoon, perhaps, in the old sense of that word, would be a more exact word to use—can best be drawn by first sketching in objectives.

The basic objective of applied research in engineering is improvement in economy. In the civil field this means economy in the financial sense; in the military field it means economy in a broader and more variable sense. This very general objective of economy leads in the case of the gas turbine to a great number of more detailed objectives, some of the more obvious of which are:—

> *Improvement in performance*—that is, getting more and more power per pound of material used. This includes:
>
> *Improvement in efficiency*—that is, getting more and more power per pound of fuel used;
>
> *Improvement in maintenance*—that is, spending less and less time and money on keeping the machine in operating trim;
>
> *Improvement in omnivorousness*—that is, adapting to the consumption of cheaper and cheaper fuels; and
>
> *Improvement in adaptability.*

The relative importance of these objectives of course varies from one application of the gas turbine to another.

In practice, it is often difficult to separate improvement in performance from improvement in efficiency, but the distinction between them is worth making. Whilst improvement in performance is frequently a consequence of improvement in efficiency, improvement in performance is sometimes achieved without improvement in efficiency. This is readily exemplified by the development of aircraft gas turbines, among which are machines which have been modified to give more power at the expense of correspondingly more fuel, the thermal efficiency remaining about the same. There are even cases in which occasional bursts of very high power are achieved with very small increases in engine weight and reduced thermal efficiency. Maintaining this distinction, which admittedly is generally *not* maintained, it can be said that increase in performance is sometimes a matter of increasing the through-put of fuel and air, with other changes, if any, consequential.

Improvement in the breed of gas turbines is most easily understood, however, in terms of thermal efficiency. High thermal efficiency is obtained by high-pressure ratio, high aerodynamic efficiency, high combustion efficiency, high gas temperature, and a high rate of heat exchange between the hot regions and the compressed air.

It by no means follows that we must aim for the highest of each in every application of the gas turbine—for example, in the aircraft gas turbine, it is not practicable to employ a heat exchanger at all; there is no room for one. But gas turbines generally demand that research must be directed towards the increase of all these quantities and in some applications, notably in power stations, it may prove practicable to aim for the highest in every department simultaneously, except that high compression and high heat exchange are to a considerable extent thermodynamically interchangeable.

A continuing objective of research and development is to increase the pressure ratio of a single multi-stage axial compressor, and another is to increase the contribution of each stage or row of blades. In pursuit of the first objective the pressure ratio of axial compressors for aircraft engines has doubled in the past 10 years; today pressure ratios of 10:1 are achieved. The contribution per stage was given a stimulus in 1948 by a research compressor of the National Gas Turbine Establishment which achieved a ratio of 4·8:1 in six stages. This set a standard which in actual engines is only now being reached. Particularly in the case of the aircraft engine, it is desirable in the interests of compactness to minimize the number of stages, and this desire is the basic reason for the study of compressors in which the air travels at supersonic speeds. At these speeds, vastly more work per stage is possible than at subsonic speeds, but practical "supersonic compressors" have not yet been achieved. It now seems that success is more likely with air-speeds in the intermediate or transonic region.

Quite as important as high pressure ratio and a small number of stages is understanding the aerodynamics of compressors so well as to be able to design them with high aerodynamic efficiency. In the subsonic speed region this understanding has apparently become widespread, for in the past few years efficiencies of 91% have become quite frequent. Greater dividends in overall efficiency, however, have recently come from improved turbine efficiency; figures of 86 or 87% are now possible, whereas 3 or 4 years ago 81 or 82% was usual.

Considerable progress has also been made with the reduction of aerodynamic losses in other parts of gas-turbine engines, and a great deal of knowledge exists on turning air round corners with minimum loss of energy.

Efficiency in combustion is now readily attained at ordinary terrestrial altitudes, and combustion problems are not generally problems of efficiency. I will mention them later. In aircraft engines, however, there is still work to do to maintain good combustion efficiency up to great heights, and the demand for compactness leads to the problem of maintaining efficiency with very high rates of heat release.

Probably the biggest dividends in performance will come from the use of higher gas temperatures. This leads to two kinds of research—material research to discover stable and practical materials, not necessarily metallic, capable of working at very high temperatures, and research in methods of cooling the turbine rotor through which the gas expands.

The progress of the modern gas turbine has been closely associated with the remarkable progress in the development of high-temperature materials, particularly steels and nickel-chrome alloys. The latter, under the proprietary name of "Nimonic," have become the most popular materials in the United Kingdom for turbine blades. It seems likely, however, that the next big step forward will be as a result of the development of methods of cooling. At the National Gas Turbine Establishment a gas turbine is running with a gas temperature of 1,140°C. Each blade is cooled by air which passes from inside the turbine rotor through numerous small-diameter holes which run radially through the blades (Fig. 40) and discharge into the main gas stream.[20]

Of the numerous other ways that have been proposed for the cooling of turbine blades one of the most interesting is that somewhat inelegantly known as "sweat cooling." The idea is to make the outer layer of the blade of a porous material so that cooling air introduced to the core of the blade diffuses out all over the surface. Blades of this kind made by sintering and by the coagulation of meshes are under test, but a turbine wheel employing them alone has not yet been run.

Blades, in compressors as well as in turbines, are liable to vibration troubles. These are most frequently forced vibrations, the elimination of which is usually a matter of intelligent and informed *ad hoc* investigation. There are more subtle oscillations, however, some of them of the general character of flutter, or coupled oscillation in flexure and torsion; and there are some, more mysterious in origin, which are still being analysed.

The function of heat exchange in a gas turbine is largely to employ heat which would otherwise be wasted to help the combustion in the job of heating the compressed air. Thus, in some designs, this air passes over surfaces heated by the passage of exhaust gas before moving into the combustion chamber. Broadly the problem is to do this kind of task compactly, and the greater the degree of heat exchange attempted, the more difficult does it become to achieve a heat exchanger of acceptable size. Some investigators lean towards the so-called regenerative type of heat exchanger in which the hot and cool fluids are passed alternately through the same passages of a matrix. This type has certainly great potential advantages but practical difficulties have so far prevented its employment in operational gas turbines, in which the recuperative type—that in which the two fluids flow in different passages, as in a motor car radiator—has been successfully used.

Improvement in maintenance is largely a matter of design and development. The gas turbine in fact shows up well in comparison with other heat engines in maintenance and in some applications its superiority more than compensates for an inferiority in efficiency to the diesel engine. Its weakest point is probably the so-called flame tube—the innermost and hottest part of the combustion chamber. There has been steady improvement of this component, however, and in that most exacting of areas, aircraft operation, the Rolls-Royce Dart gas turbine has, as I have said, achieved a life between overhauls equal to the best achieved by piston engines.

What I have called "improvement in omnivorousness" has led to wide-spread and difficult investigations into the use of a large number of different fuels. The burning of distillate fuels, as in aircraft gas turbines, is not without its problems, but they are insignificant in comparison with the difficulties of using residual oils and solid fuels under operational conditions. The drive for economy makes the liquidation of these difficulties essential. They are of two kinds. The impurities in non-distillate fuels can cause erosion of the blades and passages and a build-up of deposits, particularly between turbine blades, which interferes with the efficiency of running and may even cause stoppage (Fig. 39). Moreover, the impurities can be of a corrosive character and, even when the volume of deposit is small, can attack the turbine materials. The use of solid fuel involves the designer mostly* in the first kind of difficulty; the use of residual oil leads mostly to the second kind of difficulty.

A great deal of work has been and is being done to get rid of both kinds.[12, 21] Broadly, the cleaning of the gas produced by coal combustion is a physical problem to be solved by mechanical means. In gas turbine investigations for the Ministry of Fuel and Power, C. A. Parsons have used centrifugal separators which leave only the finest particles in the gas passing through the turbine. For the same Ministry, and in association with the work of The British Coal Utilisation Research Association, the English Electric Company have employed a combustor in which by aerodynamic means the coal granules are induced to burn on the walls of the chamber leaving a high proportion of the ash there to form a withdrawable liquid slag. Both these systems are practicable and with development should become operational. They appear preferable to electrostatic precipitation which demands very bulky apparatus, or to filters which appear to involve prohibitive loss of pressure.

The problem in the case of residual oil fuels is mainly chemical rather than physical. Many of these fuels contain vanadium and sodium salts which at gas temperatures higher than 600°C can have damaging corrosive effect. Vanadium seems to be far the more vicious. Removal of these salts from the fuel before use is not practicable, and restriction to low-temperature running is an unacceptable frustration to progress.

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* Peat ash is unpleasantly corrosive.

So long as the ash is in the solid state, physical methods such as I have mentioned for coal ash are feasible, but ash in the liquid or gaseous state consistent with operation at temperatures above about 600°C demands other remedies. Some success in its elimination, or rather neutralization, has been achieved by maintaining slightly imperfect combustion; the presence of carbon in the products of combustion tends to inhibit the action of vanadium. It seems unlikely, however, that a complete solution will be found along these lines alone. The carbon works by absorbing vanadium pentoxide in the combustion products and this blotting-paper action can be performed by certain siliceous additives to the fuel among which is kaolin. This has been demonstrated empirically in the *Auris* trials and by the C. A. Parsons research team, who have also demonstrated that the addition of either magnesia or zinc oxide reduces vanadium attack. These additives behave differently from those of the kaolin kind; they react with the vanadium to form vanadates which are not corrosive. It is possible to use both kinds of additive at once. Dr Brown and his team at the Parsons and Marine Engineering Turbine Research and Development Association found ethyl silicate more effective than magnesium bearing additives.[22, 23] The difficulty with this, as with other promising additives, is the cost. So, whilst in some circumstances kaolin, at 0·5% of the fuel cost, goes a long way towards dealing with the problem, the search continues, particularly for cheap as well as effective materials.

Meanwhile, gas turbines continue to run on residual oils. In some cases they use oil of low vanadium content; in some cases they use the known not quite 100% additives; in some cases they run at relatively low temperatures; and some follow the lead of Beznau where, with a turbine inlet temperature of 650°C, it is found possible to run for 300-hour stretches and wash the turbines with water during 20-hour stoppages.[24]

In putting down "improvement in adaptability," the last of my list of general objectives, I had in mind the important series of researches resulting from the demand to operate gas turbines at great height in aircraft. In addition to keeping the inside of the gas turbine reasonably cool to prevent it melting, it is necessary to keep the intake warm to prevent it icing up. The development of fuels which can be supplied reasonably readily, do not freeze even at −70°C, and have approximately the same desirable reluctance as paraffin to explode in an accident, is a continuing problem. The greatest problem of all is the maintenance of high performance at great height, a problem of such complexity, immensity, and importance that the leading technical countries in the world are led to vast expenditure on plants in which engines can be tested in conditions approximating to those at great heights.

The ordinary industrial gas turbine too, has its adaptability problems, but the effect of height is not serious though a number of plants operate at altitudes which enforce reductions from sea-level performance. Such is the 10,000-kW Brown Boveri set near Lima.

The special problems of adapting the gas turbine to particular sets of conditions vary widely in character. The aircraft conditions, as well as demanding high performance at height, demand compactness to a degree which excludes heat exchangers, so that high thermal efficiency must be achieved without them. Power station conditions lead to problems of long life with cheap fuels. Merchant service conditions are basically similar, though the troublesome ambience of salt is additional. The motor car seems to demand both compactness *and* a heat exchanger. Locomotive conditions tend to be national in character and in the United Kingdom emphasize the need for high efficiency at relatively low part loads, coupled with robustness and reliability; no doubt the nuclear fission and later the nuclear fusion gas turbines will develop problems all their own.

The demands of the gas turbine as an auxiliary are of course many and various, and include portability and especially quick starting. In some process work, ability to run on low-calorific-value gas is an obvious requirement.

This particular ability is demonstrated at the limit, so to speak, by the gas turbine I have mentioned, designed to use the low concentrations of methane in mine ventilator exhausts (see p. 142). It has a special form of regenerative heat exchanger conceived by the Ministry of Fuel and Power engineers and, after most interesting research investigations,[25] built by the Incandescent Heat Company.

In the United Kingdom, gas-turbine research is carried on in the heavy and aircraft industries in collaboration with a number of governmental and quasi-governmental agencies. There is a comparable programme of work in the U.S.A. and probably also in the U.S.S.R. Considerable scientific and technical contributions continue to be made in the Western European countries, in Canada, and in Australia. To do any kind of justice to this quite tremendous activity is beyond my powers if I am to keep this Paper within tolerable bounds, but a further brief reference to the British scene may be acceptable.

Supporting the British industrial effort, and particularly the aircraft engine industry with its own range of admirable research facilities, is the National Gas Turbine Establishment, originally built on the foundations of Power Jets Ltd and the gas turbine department of the Royal Aircraft Establishment. It now occupies a wonderful site in Hampshire woodland and is a combination of good taste, efficiency, and amenity as rare today in the United Kingdom as a good cuisine. The Governmental Fuel Research Station works, on a very much smaller scale, on problems of the coal-burning gas turbine, as does the British Coal Utilisation Research Association. The research establishment next in importance in the gas turbine field to the N.G.T.E., however, is PAMETRADA, to which I have already referred in connexion with residual fuel additives. This research association is concerned solely with ships' turbine machinery, though much of its work is of the utmost interest to all gas turbine technologists. For the Admiralty, an important liquid-cooled high temperature research turbine rotor is being built to extend the experimental work already done on components employing liquid-cooled blades. The use of ceramic materials for high-temperature stator blading is also under investigation. For full-scale experiment on a variety of gas-turbine problems, PAMETRADA has a 3,500 s.h.p. set which the Association designed for its own experimental purposes and on which, for example, the work already referred to on fuel additives was done.

Before these and many other investigations have taken gas turbine development much farther, a fresh set of problems of great complexity will be awaiting solution, the problems of the nuclear-energy gas turbine. From the thermodynamic point of view it will be the same as the gas turbine as we know it now, but the therms will be coming from the heat of nuclear fission (or nuclear fusion) instead of from the heat of combustion of some hydrocarbon and oxygen. The source of the new problems will be the transference of heat generated in the new way into the working fluid—the gas which passes through the rotors of the gas turbine. That gas has to be made hot without being made appreciably radio-active.

This is unlikely to be achieved with the internal combustion form of open-cycle (Figs 1 and 2) though the external combustion form of open-cycle (Fig. 3) is possible. The most obviously suitable cycle is the closed cycle (Fig. 4) which is readily adapted on paper (Fig. 41) to nuclear energy heating.

As in other closed-cycle plants, air can be used as the working fluid. Other gases of higher conductivity, such as helium are worth considering, however, since they permit more compact design. For maximum thermal efficiency, a more complex cycle (Fig. 42) is necessary and direct heat interchange between the working fluid and the radioactive fluid carrying the reactor heat is undesirable. This means the further

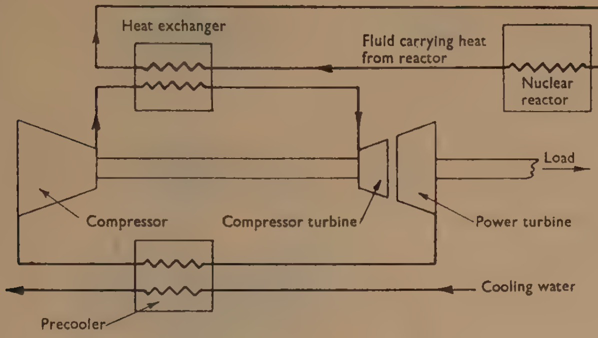


FIG. 41.—ADAPTATION OF CLOSED CYCLE TO NUCLEAR ENERGY HEATING

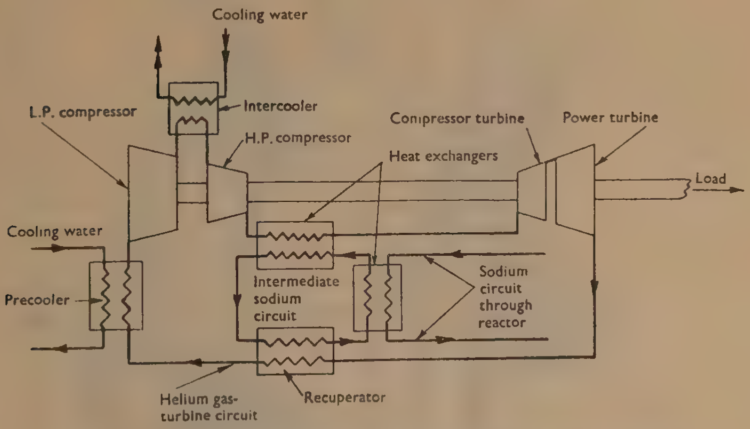
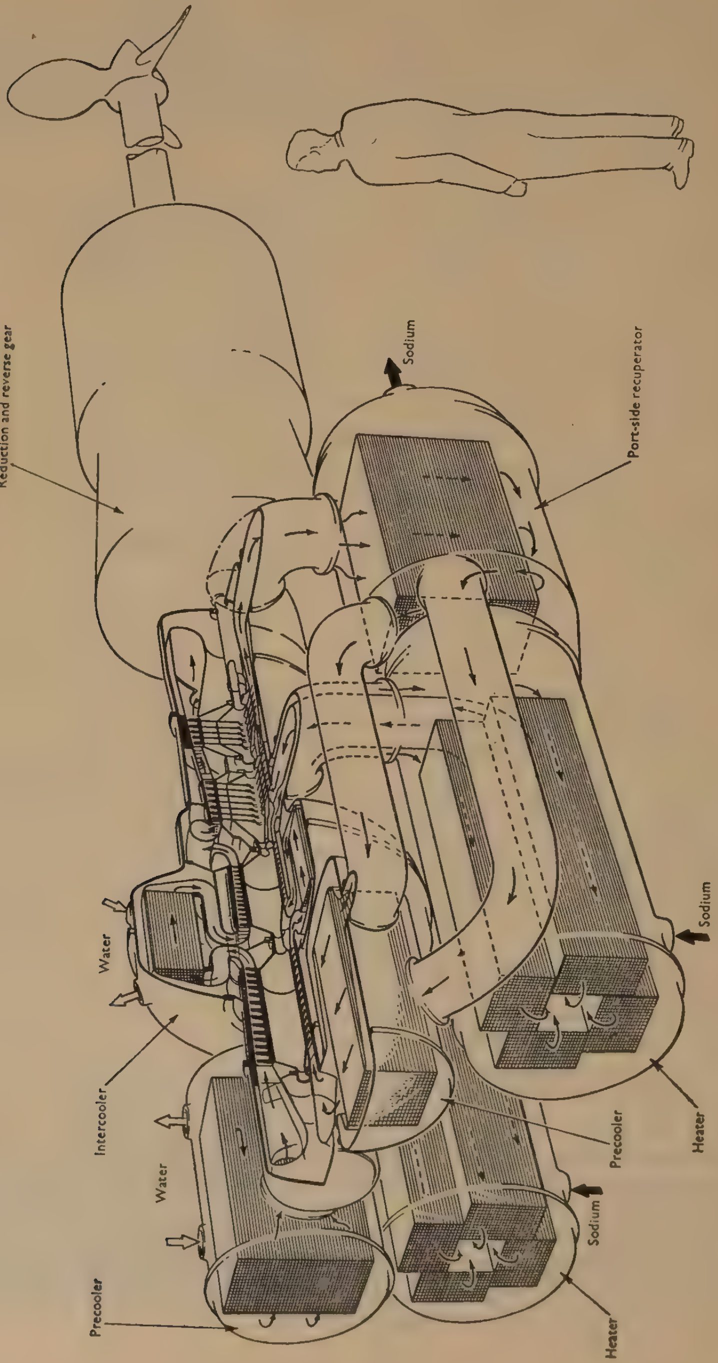


FIG. 42.—POSSIBLE NUCLEAR ENERGY GAS-TURBINE ARRANGEMENT

complication of an intermediate closed circuit, say of sodium, between the circuit, also probably of sodium, carrying the reactor heat and the working fluid, helium say, in the gas turbine.

This scheme is shown pictorially in my final diagram (Fig. 43), for which I am indebted to Messrs Rolls-Royce. The intermediate sodium circuit is invisible but, given time, the route of the helium can be traced. The nuclear reactor is invisible too, but after all, this is a lecture on gas turbines.


Reduction and reverse gear
Sodium
Port-side recuperator
Water
Intercooler
Water
Precooler
Sodium
Heater
Precooler
Sodium
Heater

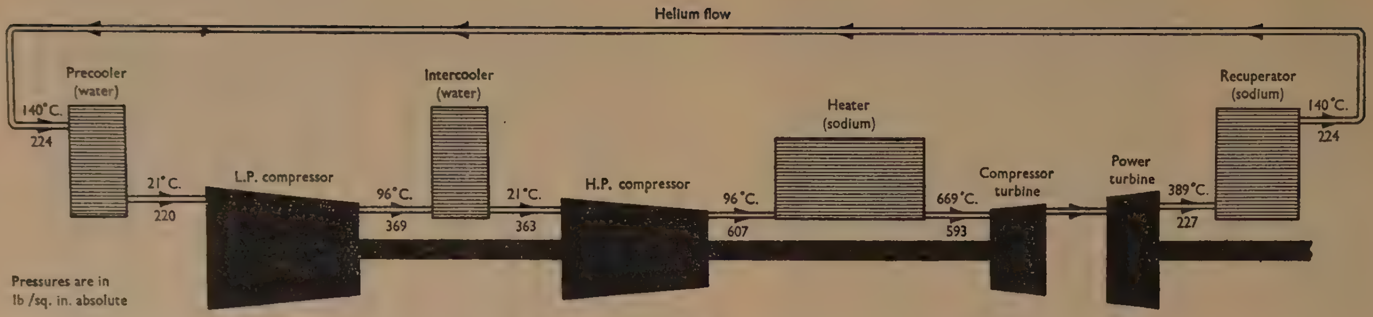


FIG. 43.—15,000 S.H.P. CLOSED-CYCLE HELIUM GAS TURBINE FOR ASSOCIATION WITH A NUCLEAR REACTOR (cf. FIG. 42)

The picture shows a marine set. But it could easily be enlarged to be a power-station set. To adapt it to aircraft or vehicular duty might be more difficult. Other and, I am assured, even more attractive, geometrical arrangements are possible, but this one I feel is adequate to foreshadow the shape of things to come.

EPILOGUE

I now come to the end of the last Paper which I intend to write on the subject of gas turbines. For the gas-turbine expert there is little new in it. For the others in this audience of many and varied talents, however, I hope not only that here and there was something they did not know before; I hope, too, that they have appreciated, if they had not already done so, the extraordinary variety of the applications of this Proteus among heat engines. Its versatility has, indeed, imposed itself as the theme of this lecture. If Hero, in his sunlit marble office in Alexandria 2,000 years ago dreamed of the future of his creation, in his wildest fancies he would not have imagined its proliferation. Even if Barber, in his possibly more prosaic study in Nuneaton, had let his imagination run riot, he could have glimpsed only a small fraction of the riches ahead. We may justly believe, however, that Parsons foresaw a great deal. Not only did he adumbrate the gas turbine in 1884; not only did he construct axial flow compressors in the early years of this century; he developed the steam turbine and saw it applied in the service of man all over the world. And had there been no steam turbine, who will dare to say that there would have been a gas turbine today?

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APPENDIX I

JOHN BARBER'S GAS TURBINE PATENT NO. 1833 OF 1791 LESS THE FORMAL PREAMBLE AND ENVOI

NOW KNOW YE, that I, the said John Barber, in compliance with the said proviso, do hereby describe the nature of the aforesaid Invention, and declare that the same is to be performed in manner following, that is to say:—

It consists of a metallic vessel called a retort, so contrived that (when heated by a circumambient fire) coal wood, oil, or any other combustible matter may be put therein, and the smoke or vapour therein collected may be brought out by a small pipe, and conveyed in a regular stream into another metallic vessel called an exploder, by means of an air pump and a compressor or regulating bellows, which pipe opposing its orifice to another similar pipe which enters the exploder (on the opposite side from the pipe which brings in the inflammable vapour from the retort) and injects by similar means a proper quantity of atmospheric or common air, causing an admixture of the two airs, which, so mixed, will take fire on application of a match or candle to the mouth of the exploder, and rush out with great rapidity in one continued stream of fire, so long as the exploder is supplied with proper quantities of the respective airs. The fluid stream is also considerably augmented both in quantity and velocity by water injected into the exploder by means of another small pipe entering therein, which water is also intended to prevent the inward pipes and the mouth of the exploder for melting by the velocity and intenseness of the issuing flame. This water, as well as the airs, are forced into the exploder by means of a pump, which in lieu of common crank carries upon the axis of one of its wheels two esses or double portions of circles, whereby a more regular motion is procured than can be done by any crank work. This engine is wrought by the steam issuing from the mouth of the exploder, and may be applied to grinding, rolling, forging, spinning, and every other mechanical operation; and the fluid stream may be injected into furnaces for smelting metallic ores, or passed out at the stern of any ship, boat, barge, or other vessel, so as by an opposing and impelling power directed against the water carrying such vessel, the vessel with its contents may be driven in any direction whatsoever.

APPENDIX II

Extract adumbrating the modern gas turbine from Sir Charles Parsons's Patent No. 6735 of 1884

Motors according to my invention are applicable to a variety of purposes, and if such an apparatus be driven, it becomes a pump, and can be used for actuating a fluid column, or producing pressure in a fluid. Such a fluid pressure producer can be combined with a multiple motor according to my invention so that the necessary motive power to drive the motor for any required purpose may be obtained from fuel or combustible gases of any kind. For this purpose I employ the pressure producer to force air or combustible gases into a close furnace of any suitable kind such as used for caloric engines into which furnace there may or may not be introduced other fuel (liquid or solid). From the furnace the products of combustion can be led, in a heated state, to the multiple motor, which they will actuate. Conveniently the pressure producer and multiple motor can be mounted on the same shaft, the former to be driven by the latter: but I do not confine myself to this arrangement of parts.

The blades should be of material that will withstand high temperatures or I may employ water or other fluid to cool the blades. This may be done by providing in the cylinders that carry the blades, channels or passages for the circulation of cooling fluid, which in the case of the rotary cylinder may be supplied through a passage or passages in the shaft carrying the cylinders.

APPENDIX III

Gas turbine collaboration committee

A friend who read this Lecture in manuscript considered that the historical part should include a reference to this Committee. I proposed its formation in the autumn of 1941 to ensure that all concerned in the development of the aircraft gas turbine should collaborate to avoid unnecessary duplication of effort and to pool ideas, testing facilities, and experience. The first meeting was on 1st November of that year, and since then the main body has met about four times a year. The members of the Committee are senior engineers from the aircraft engine firms and representatives of the Government's research and development organizations. I was the first Chairman. The continued vitality of the Committee owes a great deal to my successors, Group Captain G. E. Watt, Captain M. Luby, R.N., Air Commodore F. R. Banks, Mr R. H. Weir, the present Chairman, and the enthusiasm of the members.

Mr R. W. Mountain proposed a vote of thanks to the Lecturer, which was accorded with acclamation.

The President presented the Lecturer with an honorarium and a commemorative bronze medal, which formed part of the Parsons Memorial.

Paper No. 6095

EXPERIMENTS ON CONCRETE UNDER COMBINED BENDING AND TORSION

by
*Norman Sydney John Grassam, Ph.D., B.Sc.(Eng.), A.M.I.C.E., A.M.I.Mech.E.

(Ordered by the Council to be published in abstract form) †

Introduction

Many of the characteristics of reinforced concrete have been established under simple loading conditions but the more complex problem of its behaviour under combined stresses remains uncertain. In an attempt to provide fundamental design data an investigation was made into the strength of plain concrete members subjected to bending combined with torsion so that the problem of the effect of the reinforcement might be treated in a rational manner.

Experience with other brittle materials such as cast iron or marble under similar combinations of stress suggests that concrete might fracture at a definite maximum principal tensile stress. Another reasonable theory for brittle materials is that failure occurs when a limiting extension strain is reached. Nylander[1] in 1945 tested plain concrete specimens under compression and torsion and his results, with limitations, substantiate a maximum principal stress theory. Subsequent to the Author's research, Cowan and Armstrong[2] in 1952 tested reinforced concrete in combined bending and torsion and concluded that the addition of a small amount of bending to a reinforced concrete section increases its resistance to torsion. They also essentially concur with a maximum stress theory. Experimenters in pure torsion agree that the torque required to fracture a solid circular specimen is about 20% in excess of that calculated on the elastic theory. Similarly in pure bending the bending moment required is between 1·3 and 1·5 times that calculated by elastic formulae.

Specimens and apparatus

The principal experiments of the Author consisted of a series of twelve tests to fracture under bending and torsion applied simultaneously to plain concrete specimens having a parallel section 30 in. long and 7½ in. diameter. These were supplemented by subsidiary tests for strength on tension, compression, and unreinforced rectangular beam specimens. A series of tests[3] using electrical-resistance strain gauges was conducted to measure the surface strains induced on specimens of all these types.

The test rig for loading in combined bending and torsion is shown in Fig. 1 and was so arranged that if the complementary loads were equal the specimen was subjected to pure bending and/or to pure twisting, but not to direct forces. The

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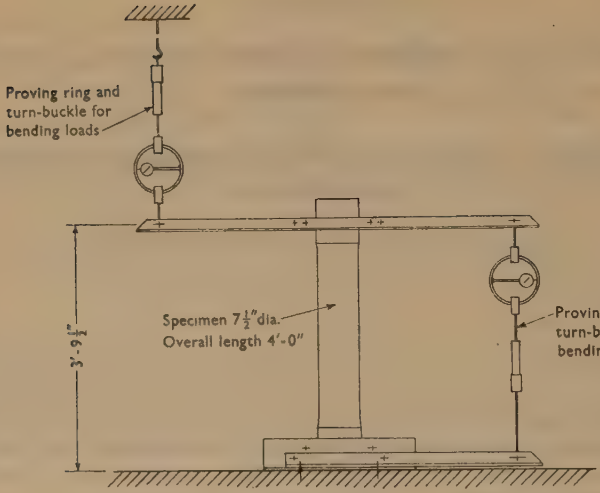
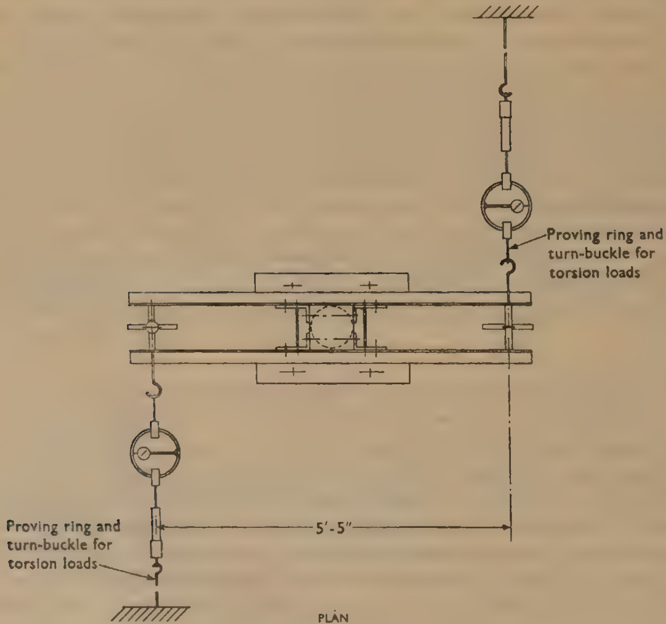
* The Author is Lecturer in Mechanical Engineering at the University of Southampton.

† The full MS. and illustrations may be seen in the Institution Library.—Sec.

[1] The references are given on p. 164.


Proving ring and turn-buckle for bending loads
Specimen 7½"dia. Overall length 4'-0"
3'-9½"
Proving ring and turn-buckle for bending loads

ELEVATION


Proving ring and turn-buckle for torsion loads
5'-5"
Proving ring and turn-buckle for torsion loads

PLAN

FIG. 1.—TESTING ARRANGEMENTS FOR COMBINED BENDING AND TORSION

loads were applied through turn-buckles and were measured by proving rings. The accuracy of measurement was ± 100 lb.-in.

The concrete was of the conventional 1:2:4 loose volume mix of rapid-hardening Portland cement "Ferrocrete," Thames Valley sand of fineness modulus 1·8, and graded crushed Thames Valley ballast of $\frac{3}{8}$-in. maximum size giving a fineness modulus for the joint aggregate of 5·0. This size was selected so that it bore a similar relation to the size of specimens tested as would normal coarse aggregate to sections used in practice. A water/cement ratio of 0·55 and water to total dry mix ratio of 7·5%, both measured by weight, was used.

Specimens were cured in water and tested wet at 28 days.

Results

The graph, Fig. 2, illustrates the results of the principal tests.

Loads were applied at a steady rate so that the total duration of a test was about 8 to 9 min. The failures were generally very sudden and without warning, along a helix.

Strain measurements were taken on specimen No. 12 during understressing tests in pure torsion and in pure bending before it was loaded to fracture under combined loads. Single resistance-strain gauges were placed in the form of rosettes and set at 0°, 45°, and 135° to the axis of the specimen at opposite ends of a diameter in the plane of bending. Readings showed the modulus of elasticity to be $2{\cdot}4 \times 10^6$ lb/sq. in. in shearing to 40% ultimate load, and $5{\cdot}06 \times 10^6$ lb/sq. in. in bending to 50% ultimate load. When torsion was applied to this specimen carrying a 50% bending moment, Hooke's law was for all practical purposes obeyed to 85% of the torque required to bring about fracture, and the shear modulus of elasticity was unaltered. The principal tensile strains immediately before the fracture could not be evaluated since local discontinuities in the material had disorganized the strain readings by that time.

Considerable scatter in the tensile test results was observed resulting principally from eccentric loading, and the average tensile strength of the concrete was taken as 330 lb/sq. in. from specimens of dumb-bell shape having a working section nominally 3 in. × 3 in. × 6 in. long. The modulus of elasticity was found to be $5{\cdot}6 \times 10^6$ lb/sq. in. to 45% ultimate load and, after making allowance for eccentric loading, the ultimate tensile strain was 90 to 95×10^{-6}.

The average compressive strength from specimens of 5-in.-diameter and 10 in. high was 3,100 lb/sq. in. The mean value of the modulus of elasticity to about 50% load was $4{\cdot}4 \times 10^6$ lb/sq. in.

Four plain beams of section 3 in. × 6 in. × 48 in. long were tested under four-point loading and gave a moment of resistance of about 7,650 lb.-in. These tests were not so satisfactory as the Author would have liked.

Analysis

Observation shows the fracture of concrete under combined bending and torsion to be by separation. Guided by the experience of others with brittle materials, and noting the general trend of the results obtained, the two common separation theories of failure are used in the following analysis—the maximum tensile stress and the maximum tensile strain hypotheses.

The concrete is assumed to be homogeneous—an assumption which is sufficiently true if the maximum size of aggregate is small compared with the dimensions of the section.

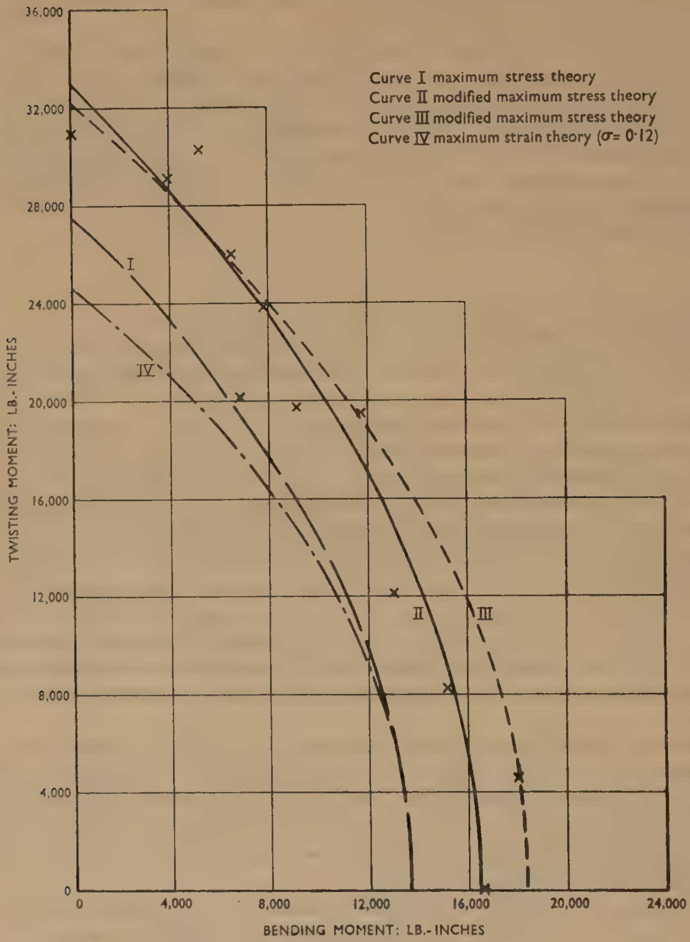


Fig. 2.—Results of combined bending and torsion tests

If Hooke's law is obeyed to fracture, then taking the tensile strength of the concrete as 330 lb/sq. in., the maximum tensile stress theory gives curve I in Fig. 2. Clearly this curve falls well inside the experimental points so that, if the figure for the tensile strength is correct, then the increased load capacity must presumably arise from the plastic behaviour of the material. Several alternatives exist for determining a theoretical curve that allows for this.

One method is to consider the stress distribution which might exist at fracture. The extreme cases are if the section remains completely elastic, i.e., the stress is proportional to radius, and secondly, if the section becomes fully plastic, i.e., the stress is uniform across the section. In pure torsion it has been shown by Marshall and Tembe[4] that to take the fracture torque to be the mean between that calculated assuming completely plasticity and assuming complete elasticity gives close agreement with experiment. The Author suggests that similarly for bending the mean between the elastic and plastic calculations be used. To derive less arbitrary factors by assuming various stress distributions is a matter of mathematical exercise, but most rational assumptions give results close to the mean. It may be noted that when this criterion is applied to the beam test results, the magnifying factor over elastic calculations is 1·25 theoretically and 1·29 experimentally. Under combined stresses it is proposed that the maximum stress theory, which can be expressed in the form $\frac{M}{M'} = 1 - \left(\frac{T}{T'}\right)^2$, holds where M' and T' are respectively the bending moment and torque calculated on the semi-plastic idea suggested and which if applied separately would cause fracture; M and T are respectively the bending moment and torque

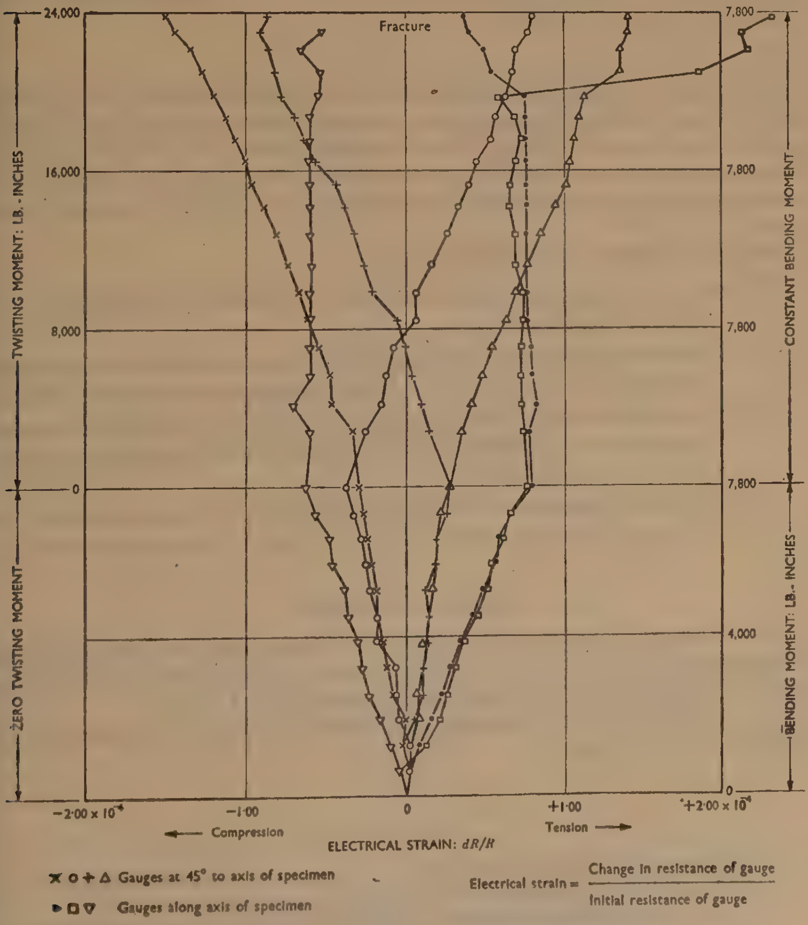


FIG. 3.—LOAD/STRAIN CURVES FOR COMBINED BENDING AND TORSION TEST

which if applied together would cause fracture. This proposition gives curve III in Fig. 2 which is a closer approximation to the experimental curve than is given by the elastic theory.

Another method is to take the design tensile stress as, say, 1·2 times the measured tensile strength. This gives curve II in Fig. 2 which is also a close approximation to the experimental curve, although the chosen factor has no theoretical justification.

The maximum strain theory using a value obtained in compression tests on this concrete of 0·12 for Poisson's ratio gives curve IV in Fig. 2. This is not greatly different from the curve given by the maximum stress theory, but it does not fit the experimental points so well. Comparison of the apparent ultimate strains measured with resistance-strain gauges shows that the tensile strain capacity of the concrete was about the same in the beam tests as in the direct tension tests, namely, a strain of 95×10^{-6}. In the combined bending and torsion test, the maximum principal tensile strain shortly before fracture was 55×10^{-6}. The strain behaviour in this test is given in detail in Fig. 3.

Conclusions

The maximum principal tensile stress theory modified to allow for some plasticity appears to give the best estimate of the criterion of fracture of plain concrete under combined bending and torsion. From a practical point of view multiplying the design stress by a factor is adequate for this modification, and for the concrete tested it is suggested that the design tensile stress be taken as 1·2 times the measured tensile stress.

General observations on the tests show that:—

(1) The torsional stiffness of the section was not altered by the application of a 50% bending moment.

(2) No increase in the torsional strength was found when a small bending moment was applied, as has been noted by experimenters on reinforced sections.

(3) The electrical-resistance strain gauge proved to be very valuable in measuring the surface deformations and often revealed faults in testing arrangements not otherwise suspected.

(4) The testing arrangement for combined bending and torsion was satisfactory and could easily be adapted to test sections of conventional shape.

ACKNOWLEDGEMENTS

The investigations described were carried out during 1945–49 in the Engineering Laboratories of the Battersea Polytechnic, London, by Dr D. Fisher and the Author, and form the subject of Ph.D. theses of the University of London.

It is regretted that other commitments prevented Dr Fisher from assisting in the preparation of the Paper.

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The Paper, which was received on the 1st June, 1955, is accompanied by three photographs and twelve sheets of drawings, from some of which the Figures in the text have been prepared.

CORRESPONDENCE
on a Paper published in Proceedings, Part I, July 1955

Paper No. 6070

" Settlement analyses of six structures in Chicago and London " †

by

Professor Alec Westley Skempton, D.Sc.(Eng.), A.M.I.C.E.,
Professor Ralph Brazelton Peck, D.C.E., M.A.S.C.E., and
Donald Hugh MacDonald, Ph.D., M.E.I.C.

Correspondence

Mr E. M. Hayward (Chief Engineer, Trollope & Colls Ltd, London) stated that whilst he fully appreciated the value of such records of settlements of buildings, a word of warning should be given to those who might be induced by such instances of occasional settlements to reduce drastically permissible foundation pressures, which were the result of the accumulation of building experience in a particular locality. There was evidence of such a tendency in London.

It had been customary 20 to 30 years ago to design foundations for a gross pressure of 3 to 4 tons/sq. ft on the upper layers of the London blue or grey Clay. Admittedly the buildings were to carry greater live loads than customary today and which might not have been realized in practice, but they showed no sign of settlement of several inches as usually forecast. Today, however, Local Authorities were reluctant to permit more than about 2½ tons/sq. ft on that clay. The Institution Code of Practice No. 4 on Foundations recommended pressures lower than those previously adopted. Skempton [29] had suggested a pressure of about 2 to 2½ tons/sq. ft on London Clay for modern buildings of ten to twelve storeys. What practical justification was there for such a reduction? Were engineers being unduly influenced by the publication of such articles describing a ¾-in. settlement of a small building at Elstree under a pressure of about 1 ton/sq. ft on firm brown clay? Had the Authors considered any other reasons for that insignificant settlement? Had they checked any other buildings in the same locality?

There were numerous buildings in London designed for 2 tons/sq. ft on similar brown clay, which had been behaving quite satisfactorily for many years.

Mr Hayward referred to an office building which had five storeys and was steel-framed with foundations designed for 3¾ tons/sq. ft. It had been built about 16 years ago and was flanked on two sides by old brick buildings built about 50 to 100 years ago. At two levels stone string courses had been bonded to the adjacent older buildings. Those bondings had not cracked. Theoretically the settlement of the flanking buildings should have reached equilibrium but the new building should have settled about 1 to 3 in. It evidently had not settled and a walk round the City of London would provide numerous similar examples of new buildings bonded to older buildings or built tight against them without any signs of distress. Mr Hayward had knowledge of many important buildings constructed in the London area during the past 30 years. They had been built on London brown, grey, and blue Clay with gross foundation pressures between 2 and 4 tons/sq. ft. No serious settlements had occurred. Could the Authors give instances of modern buildings in London which had settled appreciably?

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† Proc. Instn Civ. Engrs, Part I, vol. 4, p. 525 (July 1955).

Many engineers could give examples of new extensions to existing buildings. In his experience the floors of the new buildings had invariably been made level with the existing buildings and no allowance was made for relative settlements. No noticeable settlement had occurred in such cases. What did the Authors think was the reason for that apparent discrepancy between theory and practice?

In the words of Terzaghi "it is obvious that the results of a settlement computation are not even approximately correct unless the assumed hydraulic boundary conditions are in accordance with the drainage conditions in the field." If the settlement constants were computed from a laboratory experiment with the sample held between porous plates, Mr Hayward suggested that that did not represent the true hydraulic boundary conditions of the foundations of a new building adjacent to or surrounded by existing buildings. If the surrounding clay was already stressed by existing buildings, the pore-water under the new building could not be forced into the pores of the adjoining sites unless the pore-water pressure under the new building was appreciably greater than the pore-water pressure in the clay under the adjacent buildings. Was that a possible explanation of why new buildings in built-up areas did not appear to settle so much as forecast?

Tattersall, Wakeling, and Ward [30] had stated:—

> "Undrained triaxial compression tests made on pairs of 1½-in.-dia. samples extracted from 4-in.-dia. cores taken from 80 to 100 ft depth were generally less than one-third the strength of samples obtained from the tunnel and trimmed by hand. Even at a depth of only 40 ft at this site the tests on cored samples probably underestimated the strength considerably. The material was too hard for core samples to be undisturbed."

Meyerhof and Murdock [31] had observed that there was significant decrease in the compressive strength of samples which had been stored in tubes for several days. They stated that:—

> "This decrease in strength, which amounted up to about one half of the original strength, was particularly noticeable for the stiffer and harder material and may be explained by the gradual opening of fissures of the clay after sampling."

The usual laboratory triaxial tests obviously gave results for London Clays lower than the real strengths of the clays in their undisturbed state. The factor of safety of 2½ suggested by Skempton [29] and the Code of Practice was probably too high and there seemed insufficient justification for taking only 2 to 2½ tons/sq. ft pressure on London Blue Clay. Probably the laboratory estimates of elastic modulus and coefficients of consolidation were also unreal, which might account for the fact that buildings in London generally did not settle so much as expected from theoretical computations.

It would be remarkable if a simplification such as equation (1) gave anything but an approximation to the correct answer to problems connected with such a heterogeneous anisotropic material as soil. In the synopsis the Authors had bravely stated that "settlements can be computed with errors not exceeding the limits of −30% and +50%." In Table 2 by an unfortunate selection of case histories those wide limits were reduced to an average "only 9% greater than the observed value." In view of the many assumptions governing the basic equation, it would be almost a fluke if the correct answer was given to within 9%. It might be ungenerous, but he could not help wondering whether it was possible for a little wishful thinking to choose or adjust the constants of the computations to give an anticipated result? Engineers were familiar with the use of dynamic pile-driving formulae alleged to give the static load bearing capacity of a pile and in which small alterations in the constants would give the anticipated or desired answer.

He welcomed the recording of factual observations as given in the Paper, but until far more than those few case histories were available it was advisable to accept Terzaghi's warning that "the computation of the rate of settlement has the character of a crude estimate." The Authors wisely acknowledged the limitations of their theories in their statement that "Theoretical considerations of this problem are difficult and have not yet been completed." Mr Hayward suggested that those few records of settlements were no valid reason for lowering the permissible foundation pressures, which had been established as safe and satisfactory by long experience of building in any locality.

Mr J. K. Alderman (Lecturer in Civil Engineering, Manchester University) observed that, on examining the time/settlement curves for the six structures analysed, it was difficult to understand why the Authors advocated the use of the equation:

$$\rho_f = \int_0^z m_v \,.\, \Delta\sigma \,.\, dz \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

for the net final settlement of a foundation, and did not consider the use of the more fundamental equation for the net final settlement:

$$\rho_{\text{final}} = \rho_i + \rho_c \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

where ρ_i denoted the immediate settlement $= q_n \,.\, B\dfrac{(1 + \nu^2)}{E} I\rho$

and ρ_c „ the consolidation settlement $= \displaystyle\int_0^z m_v \,.\, \Delta\sigma \,.\, dz \; (= \rho_f)$,

together with the corresponding equation for the net settlement ρ_t at any time, t,

$$\rho_t = \rho_i + U\rho_c \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (3)$$

The time/settlement curves for the six structures had been calculated using equation (3) and it was found that for all the Chicago structures the recalculated curves were closer to the observed time/settlement curves than any of the calculated values given by the Authors (see Fig. 8). For the structure of the Fire Testing Station, Elstree, the recalculated curve overestimated the settlement by about 25%, but it should be noted that the field settlements had been observed only for 4 years and were still on the linear part of the settlement/$\sqrt{\text{time}}$ curve. It was therefore possible that after a further time the observed settlements might cross over the Authors' calculated curve.

It had been found impossible to recalculate the time/settlement curves for Waterloo and Chelsea Bridges since the divergence with time between the two methods of calculation could not be understood. Using the previously defined symbols then:

$$\begin{aligned}\rho_t \text{ (calculated by Authors)} - \rho_t \text{ (conventional)} &= \{\rho_i + U(\rho_c - \rho_i)\} - \{U \,.\, \rho_c\}\\ &= \rho_i(1 - U)\end{aligned}$$

The difference between the two calculated values should be convergent as the time increased. Figs 3 and 4 in the Paper, however, showed the two curves were divergent with time.

The method of calculation for the net settlement at any time t suggested by the Authors and given in the equation:

$$\rho_t = \rho_i + U(\rho_c - \rho_i) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (4)$$

also broke down if $\rho_i > \rho_c$. The Authors had not mentioned that possibility and in fact stated that "the net final settlement is typically between two and six times the settlement at the end of construction." It had been found, however, during settlement calculations for the analysis of structures on glacial clays in Lancashire that $\rho_i > \rho_c$ and that the settlement at the end of construction was about 60% of the net final settlement (Fig. 9). No full-scale site observations were yet available to confirm the above analysis, but model

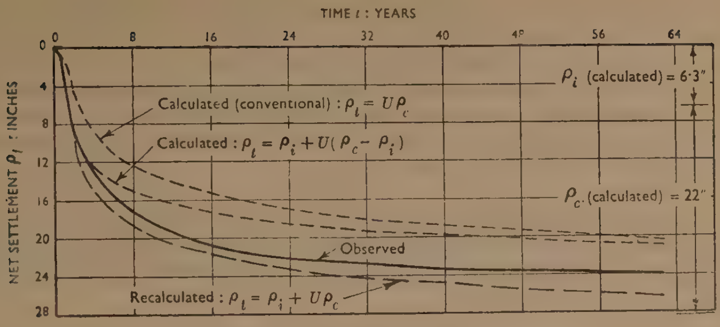

Fig. 8.—Auditorium tower, Chicago

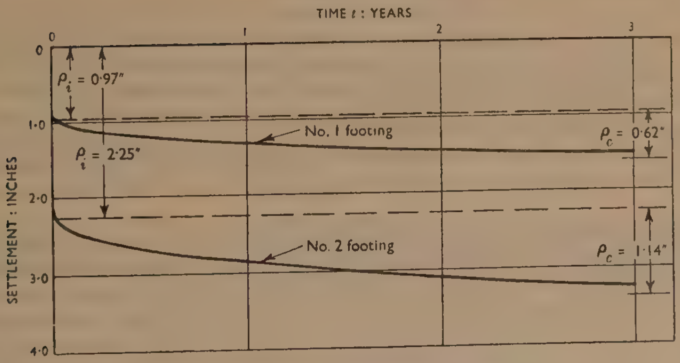

Fig. 9.—Settlement of foundations on glacial clay

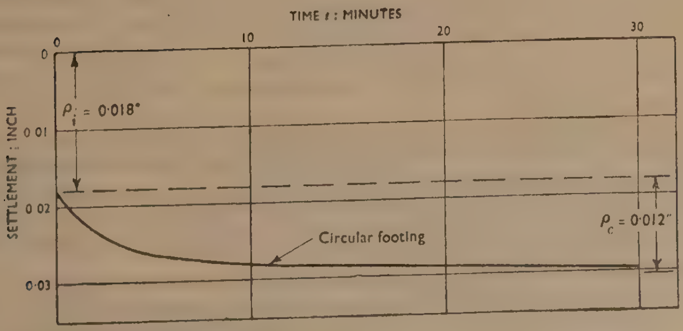

Fig. 10.—Settlement of model footings

footing tests on undisturbed samples from the field had shown that the immediate settlement ρ_i was greater than the consolidation settlement. Fig. 10 showed the time/settlement curve for a model footing test on a typical undisturbed sample from the site analysed in Fig. 9, from which it could be seen that in both cases the ratio $\rho_i/\rho_c = 1{\cdot}5 - 2{\cdot}0$. If that value was substituted in equation (4) the settlement at any time t was found to decrease as the time increased. It was therefore suggested that until full-scale site observations were available equations (2) and (3) should be used for estimating the settlement of foundations in Lancashire glacial clays.

Professor G. G. Meyerhof (Head, Department of Civil Engineering, Nova Scotia Technical College, Halifax, N.S., Canada) considered that the net settlement at any time could, in addition to the two methods mentioned by the Authors (equations 4 and 5) also be estimated from the equation

$$\rho_t = \rho_i + U\rho_{\text{final}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where ρ_i denoted the net immediate settlement (equation 2), ρ_{final} the final consolidation settlement (equation 1), and U the degree of consolidation.

That equation was based on a theoretical approach,[32] whereas the Authors' equation 5) was of a semi-empirical nature.

Using equation (6) the calculated rates of settlement agreed somewhat better with the observed movements of the six structures, especially at Chicago, than did the Authors' equation (5). Whilst at the end of the construction equations (5) and (6) gave practically the same result for the net settlement, equation (6) overestimated the final net settlement, as had been mentioned by the Authors, the mean ratio of calculated/observed final settlement for the six structures being 1·4 (using equation 6) compared with 1·1 (using equation 5). On balance it might therefore be concluded from an analysis of the present cases that there was little to choose between equations (5) and (6) and both equations were definitely superior to the conventional equation (4) as was stated by the Authors.

Professor Skempton, in reply to Mr Hayward, wished to make it clear that the Paper did not deal with allowable settlements of buildings, but with the records of six structures for which both settlement calculations and observations were available in some detail. And those records had shown that the method of settlement computation at present in use gave results which were substantially correct. Moreover, in another Paper,[33] it had been shown that, so far as final settlements were concerned, that conclusion was valid for a total of twenty structures in various parts of the world. Much had still to be learnt, however, and Mr Hayward could be assured that work was actively proceeding to improve the methods of settlement calculation.

Mr Hayward evidently felt that, in London Clay, soil tests were unreliable and computations severely overestimated settlements. But for the three structures on that Clay, described in the Paper, the calculated final settlement was, on the average, only 33% in excess of the actual value; and at the time of the latest recorded settlement the error was less than 20%. At the new Government Offices in Whitehall the maximum settlement two years after completion of the building was 1·4 in., compared with a calculated value of 1·5 in.[34]

In the past it seemed often to have been thought that because there were no cracks in a building, it had not settled or, at least, that the settlements were not appreciable. Yet there were many buildings on clay which had undergone maximum settlements of 3 or 4 in. but which had shown no signs of any cracking. Therefore it was unjustifiable to assume that little if any settlement had occurred simply because a building was not cracked or visibly distorted.

It was true, as Mr Hayward had pointed out, that there were many buildings in London erected immediately alongside older buildings, with string courses lined up with, or even bonded into, the adjacent structures; and yet no visible differential settlement had occurred. A probable explanation was simply that the clay under the new building was pre-composed by the old building which had been demolished to make way for the new structure which, in many cases, imposed no load on the clay additional to that which had been there before. Indeed, with modern structures, the weight might be less. Therefore the settlement of the new building would be very small and comparable in magnitude to the elastic recovery of the clay when the old building had been removed.

With regard to the allowable pressures on London Clay, although that matter was outside the scope of the Paper, Professor Skempton considered that his recent statement to the effect that 2 to 2½ tons/sq. ft was suitable for the footings of modern blocks of flats represented no departure from accepted practice. Those buildings usually had no basements, and the footings were typically about 8 or 10 ft below ground surface. In the outer regions of London that depth was, in most cases, still well within the zone of the brown clay, on which higher pressures had rarely been used. In fact, Mr Hayward had implied that a pressure of 2 tons/sq. ft was common on the brown clay. The settlement of ¾ inch at Elstree under a pressure of 1 ton/sq. ft could hardly have led to a decision to reduce the allowable pressures on the brown clay, since, as Mr Hayward himself had pointed out, that settlement was insignificant.

In reply to Mr Alderman and Professor Meyerhof there could be no doubt that the equations

$$\rho_{\text{final}} = \rho_i + \rho_c$$

and

$$\rho_t = P_i + U \cdot \rho_c$$

were correct. The problem, however, was how to determine ρ_c. Mr Alderman and Professor Meyerhof had both suggested that

$$\rho_c = \int m_v \cdot \Delta\sigma \cdot d_z$$

but, as the Authors had stated on p. 528 of the Paper, that assumption was not generally true. In fact, it was more nearly correct to say that

$$\rho_{\text{final}} = \rho_i + \rho_c = \int m_v \cdot \Delta\sigma \cdot d_z$$

The oedometer test measured the compressibility of the clay under increasing vertical stress with no lateral movement. In contrast, the ground under a foundation was subjected to shear stress (as well as vertical stresses); there were lateral movements and pore pressures set up when the foundation load was first applied were, in general, very different from those set up when the load was applied to an oedometer specimen.

A rigorous treatment of the problem would require complex triaxial testing in two stages, firstly to measure the properties of the clay relevant to immediate settlement, followed, secondly, by measurement of the subsequent volume changes during consolidation. To relate the problem to present technique, however, the following treatment might be helpful.

If the settlement, as determined from the oedometer test was denoted by ρ_{oed} where, as usual, $\rho_{oed} = \int m_v \cdot \Delta\sigma . d_z$ and if the true consolidation settlement $\rho_c = S . \rho_{oed}$ where S was a factor depending, primarily, on the compressibility ratio of the clay[35] which, in its turn, depended chiefly on the geological history of the clay, then:

$$\rho_{\text{final}} = \rho_i + S \cdot \rho_{oed}$$

and

$$\rho_t = \rho_i + U . S \cdot \rho_{oed}$$

Theoretical considerations showed that for normally-consolidated clays S was a little less than 1·0, whilst for over-consolidated clays the value was roughly 0·5, or even less if the clay had been heavily over-consolidated.

Those conclusions were in broad agreement with the observations reported in the Paper; and the "recalculated" curve, plotted in Fig. 8 by Mr Alderman for the Auditorium Tower, was an example of the case $S = 1$ for normally consolidated clay, as was the analysis by Professor Meyerhof.[32] Dr L. Bjerrum and Professor Skempton were hoping to publish the theoretical work within the coming year. But, for the present, the widely adopted assumption that $\rho_{\text{final}} = \rho_{oed}$ could continue to be used, as an approximation, since it was known to give results not differing from field observations by more than 50% in the great majority of clays and involved only the simplest testing and computational procedure. The Authors were glad that Professor Meyerhof agreed with their conclusion that the conventional method of calculation, as represented by equation (4), was definitely inferior to the procedure used in the Paper.

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ELECTION OF ASSOCIATE MEMBERS

The Council at their meetings on the 13th December, 1955, and the 17th January, 1956, in accordance with By-law 14, declared that the following had been duly elected as Associate Members:—

Abson, Alec Lionel, B.Sc. (*Leeds*).
Adam, Iian Graham, B.Sc. (*Glasgow*).
Adams, Hugh Campbell, B.Sc.(Eng.) (*London*).
Allison, Harry Edwin, B.Sc. (*Birmingham*).
Anandarajah Kanagasabapathy, B.Sc. (Eng.) (*London*), Grad.I.C.E.
Aspinall, Gilbert, B.Sc. (*Leeds*).
Backler, Albert Peter, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Baker, Richard Anthony.
Ball, John Geoffrey, B.Sc. (*Leeds*).
Barnwell, Francis, B.Eng. (*Liverpool*), Grad.I.C.E.
Bartley, Maurice Albert Frederick, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Bastable, John Ernest.
Batty, Frank Sudbury, M.Eng. (*Liverpool*).
Beavan, Gordon Cornelius Granville, B.Sc. (*Bristol*), Grad.I.C.E.
Bell, Robin Francis Weare, B.Sc. (*Bristol*), Stud.I.C.E.
Bellis, Alfred Douglas Macdonald, B.Sc. (*Durham*), Grad.I.C.E.
Bentley, Ivor, B.Sc. (*Manchester*), Grad. I.C.E.
Biggart, Graham Ure, B.Sc. (*Glasgow*).
Billings, Daniel Gwynne, B.Sc. (*Wales*).
Biszewski, Edmund.
Black, Albert.
Boardman, John Douglas, B.Sc.Tech. (*Manchester*).
Booker, Wilfred Henry.
Botwood, Percy Morgan, Grad.I.C.E.
Bowles, Fred.
Bowman, Robert Cairns, B.Sc. (*Glasgow*), Grad.I.C.E.
Boustany, Faiz Wadie, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Bratty, Stuart Thompson, B.Sc. (*Belfast*), Stud.I.C.E.
Braybrooks, Robert John Escott.
Buswell, John Charles, B.Sc.(Eng.) (*London*).
Butters, Edward Albert.
Caillard, Bernard Esmond Penalver, B.A. (*Cantab.*).
Caldwell, Maurice Joseph, B.Sc. (*Belfast*), Stud.I.C.E.
Candlish, Thomas Tait, B.Sc. (*Glasgow*), Grad.I.C.E.
Casson, Walter, B.Sc.Tech. (*Manchester*).
Chalmer, Thomas Campbell, B.E. (*Sydney*).
Chambers, Alan.
Chambers, Eric Hicks Terence, B.Sc. (*Belfast*).
Charles, Alfred Louis.
Cherry, Robert Denis, B.Sc.Tech. (*Man*chester), Grad.I.C.E.
Christie, Angus Webster.
Christopher, Paul, B.Sc.(Eng.) (*London*).
Clarke, James Dennis, B.Sc. (*Durham*), Grad.I.C.E.
Clarke, Royston Thomas John, B.Sc. (Eng.) (*London*).
Cleaver, Kenneth.
Cochrane, Stanley Robert, M.Sc. (*Belfast*), Grad.I.C.E.
Cooke, Brian Joseph, B.Sc. (*Durham*).
Cope, Geoffrey Hulme, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Cornish, Dennis Radford.
Corns, Samuel Maurice.
Crocker, Peter, Grad. I.C.E.
Crosbie, Edgar, B.Sc. (*Edinburgh*).
Crozier, Antony Caton, B.Eng. (*Liverpool*), Stud.I.C.E.
Cummins, James Ashley Aylmer.
David, Geoffrey Boverton.
Davidson, James, B.Sc. (*Glasgow*).
Davidson, James Young, B.Sc. (*St Andrews*), Grad.I.C.E.
Davies, William Rhys, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Delany, Edward Patrick, B.Sc. (*Glasgow*), Stud.I.C.E.
Donaldson, Alistair Hamo.
Drew, Gordon Lionel Kingsley.
Dunn, Thomas, B.Sc. (*Durham*).
Dyson, Claude, B.Sc.(Eng.) (*London*).
Eagles, Harold Edwin Merion, B.Sc. (Eng.) (*London*).
Edwards, Douglas Charman, B.Sc. (*Birmingham*), Grad.I.C.E.
Erskine-Murray, Patrick Elibank, M.A. (*Cantab.*).
Fernandez, John Anthony, M.Sc. (Eng.) (*London*)
Field, James, B.Sc.(Eng.) (*London*).

Forster, Brian Wyndham.
Fowler, Richard James, B.Sc.(Eng.) (*London*).
Friggens, James Bodilly.
Fuge, Thomas Dymond, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Furness, Peter Bryan, Stud.I.C.E.
Garrod, Arthur Douglas, B.Sc.(Eng.) (*London*).
Garside, David Hughes.
Garvie, Denis Scott, B.Sc. (*Aberdeen*).
Gedge, David Gregory, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Gledhill, Geoffrey Maude, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Goldie, Robert.
Goodwin, Stanley Albert, B.Sc.(Eng.) (*London*).
Gray, Alexander, B.Sc. (*Aberdeen*), Grad.I.C.E.
Greenhouse, William James, B.Sc.(Eng.) (*London*).
Greenlees, Frank Millar, B.Sc. (*Glasgow*).
Griffin, James.
Griffin, William John, Grad.I.C.E.
Grisdale, Thomas Frederick, B.Sc. (Eng.) (*London*).
Hall, Charles Wilson, B.E. (*New Zealand*), Grad.I.C.E.
Hamilton, Alexander McEwan Douglas, Grad.I.C.E.
Harmer, Stanley John.
Hartman, Wlodzimierz Adam.
Harvey, Joseph Thomas, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Hayes, John David, B.Sc. (*Wales*), Stud. I.C.E.
Hebditch,J ohn Anstey, B.A. (*Cantab.*), Grad.I.C.E.
Henderson, Philip Russell, B.Sc. (*Birmingham*).
Hodgkinson, Arthur, B.Sc.Tech. (*Manchester*), Grad.I.C.E.
Holman, Peter Derek, Grad.I.C.E.
Holmes, Malcolm, B.Sc. (*Leeds*), Grad. I.C.E.
Hossack, Adam Wilkie, B.Sc. (*Edinburgh*), Grad.I.C.E.
Houghton, Philip Richard Stephen, B.Sc. (*Birmingham*), Grad.I.C.E.
Howse, Henry Desmond.
Humby, Eric James, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Hyett, Lemuel Henry, B.Sc.(Eng.) (*London*).
Johnson, Albert Edward, B.Sc.(Eng.) (*London*).
Johnston, Ian Desmond, B.Sc. (*Glasgow*).
Jolliffe, George Alan Patrick.
Kemp, George, B.Sc. (*Birmingham*), Grad. I.C.E.
Kennedy, Geoffrey Farrer, M.A. (*Cantab.*).
Khalafalla, Ahmed Omer.
Kirk, Joseph Dunn, B.Sc. (*Belfast*).
Langdon, John, B.Sc.Tech. (*Manchester*), Grad.I.C.E.
Lau Foo Sun, B.Sc.(Eng.) (*London*), Grad. I.C.E.
Law, Keith Knapton, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Lea, William Nigel, B.Sc. (*Birmingham*), Grad.I.C.E.
Levesley, Gerald.
Levy, Mousa Sassoon, Grad.I.C.E.
Lewis, Francis Hutcheson, B.Sc. (*Belfast*).
Lewis, William Murray, M.A. (*Cantab.*), Grad.I.C.E.
Lumb, Peter, M.Sc.(Eng.) (*London*), Grad. I.C.E.
McCosh, Andrew Kirkwood, B.A. (*Cantab.*).
Macdonald, Alastair Kinross, M.A. (*Cantab.*), Grad.I.C.E.
McGuffie, Gordon William, B.Sc. (*Manchester*), Grad.I.C.E.
McIntyre, John Alexander, Grad.I.C.E.
Mackay, Ian, B.Sc. (*Aberdeen*).
McKay, Donald Crowther, B.Sc. (*Wales*), Grad.I.C.E.
Mack Smith, Robin, M.A. (*Cantab.*), Stud. I.C.E.
McMullan, Joseph Hugh Gordon, B.Sc. (*Belfast*).
McNamara, Michael Martin, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Mander, Reginald Frederick, Grad. I.C.E.
Marland, Kenneth Charles.
Marsh, Peter Louis Anthony, B.Sc. (*Cape Town*), Grad.I.C.E.
Maskell, Alan David, B.Sc.(Eng.) (*Natal*), Grad.I.C.E.
Merry, Sean, B.E., B.Sc. (*National*).
Mettam, John Dorrington, M.A. (*Cantab.*), Stud.I.C.E.
Millar, James Angus, B.Sc. (*Belfast*), Grad.I.C.E.
Miller, Peter Temple Steuart, B.Sc. (*Cape Town*), Grad.I.C.E.
Millington, James Edward.
Mills, Alan Thompson, B.Sc. (*Durham*), Grad.I.C.E.
Moller, Kristen, B.Sc.(Eng.) (*London*).
Morice, Peter Beaumont, B.Sc. (*Bristol*), Grad.I.C.E.
Morris, Raymond Stanley, Grad.I.C.E.
Munro, James.
Munro, Ian Thomas, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Myles, Alec Frater, B.Sc.(Eng.) (*London*), Grad.I.C.E.

Naraine, Shiw Sahal, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Nelson, Kenneth Davies, B.Sc. (*Wales*).
Nicholson, Thomas Hedley, B.Sc. (*Durham*), Grad.I.C.E.
Norman, John Michael, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Norris, John Walton.
Orr, Samuel, B.Sc. (*Belfast*), Grad.I.C.E.
Owen, John Benjamin Brynmor, D.Sc. (*Oxon.*), M.Sc. (*Wales*).
Parsons, Geoffrey Frank, Grad.I.C.E.
Patrick, John George, B.Sc.(Eng.) (*London*), Stud.I.C.E.
Payne, Michael Roger, B.Sc. (*Bristol*).
Percival, John Adrian, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Phillips, James Roland, B.Sc.(Eng.) (*London*).
Phillips, John William, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Pigott, Alan Charles John, M.Sc.Tech. (*Manchester*).
Pillai, Alfred, B.Sc.(Eng.) (*London*).
Pittaway, John Lionel Greville, M.A. (*Cantab.*).
Pollard, James Watts, B.Sc. (*Manchester*).
Potter, Frank Higton, B.Sc.Tech. (*Manchester*).
Preece, Robert Edward, B.Sc. (*Birmingham*).
Reekie, Robert, B.Sc. (*St Andrews*), Stud.I.C.E.
Ritchie, Alistair George Baldwin, B.Sc. (*Glasgow*), Grad.I.C.E.
Robinson, Daniel Bird, B.Sc. (*Durham*).
Robinson, George, B.Sc. (*Durham*).
Roomes, Francis Godfrey, B.Sc.(Eng.) (*London*).
Sadler, John Vernon, B.Sc.(Eng.) (*London*).
Saverymuttu, Hector Sebastian, B.Sc. (Eng.) (*London*).
Shackleton, Henry Leslie, B.Sc. (*Leeds*).
Shaw, Laurence Sime, B.Sc. (*St Andrews*), Grad.I.C.E.
Shortall, Harold Rosslyn, B.A., B.A.I. (*Dublin*).
Sinclair, Robert Bryson, B.Sc. (*St Andrews*).
Smethurst, John Derwent, B.Sc. (*Bristol*), Stud.I.C.E.
Smyth, John, B.Sc. (*Belfast*), Grad.I.C.E.
Stead, Peter Alan, B.Sc. (*Leeds*).
Stephens, George Anthony, B.Sc.(Eng.) (*London*).
Stephenson, David Usher Hamilton.
Still, Robert, B.Sc. (*Aberdeen*).
Strickland, Ronald MacKenzie, B.E. (*W. Australia*), Grad.I.C.E.
Tavener-Smith, Norman Wilson, B.Eng. (*Liverpool*), Grad.I.C.E.
Taylor, Roy, B.Sc. (*Nottingham*), Grad.I.C.E.
Thatcher, Eric Frank, Stud.I.C.E.
Thomson, John Wallace, B.E. (*Queensland*).
Thorne, Robert Basil.
Tsou Huan-Hsin, B.S. (*Lingnan*).
Turnbull, Malcolm Mackenzie.
Turner, Henry George, Grad.I.C.E.
Turner, John Arthur, B.Sc. (*Durham*).
Vail, Anthony John, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Varley, William Raymond, B.Sc.Tech. (*Manchester*), Grad.I.C.E.
Verma, Ramjee Prasad, M.Sc.(Eng.) (*London*).
Virtue, John Clive, B.E. (*Tasmania*).
Waelend, John Peter, B.Sc.(Eng.) (*London*).
Wardroper, James Herbert, B.Sc.(Eng.) (*London*), Grad.I.C.E.
Watkins, William Gwyn, B.Sc. (*Wales*).
Watson, Donald Frederick, B.Sc. (*Birmingham*).
Watson, Victor Ernest, B.Sc. (*Glasgow*).
Weathersead, Donald Cameron, B.Sc. (Eng.) (*London*).
Weeks, David Parkes, B.Sc.(Eng.) (*London*).
Westall, John Butterworth, B.Sc. Tech. (*Manchester*).
Whelan, Michael Noel, B.Eng. (*Liverpool*), Grad.I.C.E.
White, Graham Charles Sidney, B.E. (*New Zealand*), Grad.I.C.E.
Whitley, Ian Hunter, Stud.I.C.E.
Whitton, Thomas Edward, B.Sc. (*Durham*), Stud.I.C.E.
Whitworth, Andrew Allen, M.A. (*Oxon.*), Grad.I.C.E.
Wigmore, Peter Roland Sydney, B.Sc. (Eng.) (*London*).
Williams, Thomas Eifion Hopkins, M.Sc., Ph.D. (*Wales*).
Williamson, Frederick Jack, B.Sc. (Eng.) (*London*), Grad.I.C.E.
Wills, Douglas Albert, Grad.I.C.E.
Wilson, Norman Angus, Stud.I.C.E.
Wong, Chung-ngar, Peter, B.Sc. (*Hong Kong*).
Wood, Robert Edward, B.Sc.(Eng.) (*London*).
Wright, William Lomax, B.Sc.Tech. (*Manchester*).
Wyatt, John Alfred, B.Sc.Tech. (*Manchester*).
Youngs, Derek Martin.

DEATHS

It is with deep regret that intimation of the deaths of the following has been received.

Members

Lt.-Col. Graham Townsend Bennett, O.B.E., B.Sc. (E. 1922, T. 1945) (*former Member of Council*).
George Rudd Collinson, B.Sc.(Eng.) (E. 1911, T. 1922).
Ernest Hone Ford, O.B.E. (E. 1911, T. 1931).
William Hubert Kirby, M.C. (E. 1911, T. 1939).
Ernest Edward Morgan (E. 1935).
Sir Richard Augustine Studdert Redmayne, K.C.B., M.Sc. (E. 1908) (*Past-President*).
Bernard Alexander Rice, B.A., B.A.I. (E. 1929, T. 1946).
Harry Watson Smith, C.B.E. (E. 1942).
Albert Tulip, M.Sc. (E. 1928).

Associate Members

Francis Thomas Hilton Bradley, B.E. (E. 1948).
George Gladman Braid (E. 1900).
Edward Arthur Evans, B.Sc. (E. 1910).
Walter Paul Frederick Fanghaenel, B.Sc.(Eng.) (E. 1915).
Robert Gilchrist (E. 1908).
Arthur Stanley Glover, M.C., B.Sc. (E. 1918).
William Faulkner Lowe, M.C., M.Eng. (E. 1934).
John Halliwell Mawson, M.C., B.Sc. (E. 1914).
Cecil Edward Mays (E. 1919).
Arthur David Clore Parsons, B.A. (E. 1909).
Ernest Lancelot Pledger (E. 1910).
Anthony Bean Saunders-Jacobs, B.Sc.(Eng.) (E. 1929).
James Henry Segrave (E. 1909).
Thomas Reginald Sewell, M.C., B.Sc. (E. 1911).
Arthur Lee Whitwam, B.A. (E. 1940).
Robert Alfred Wilkinson (E. 1879).

Graduates

Gordon Derek Hill (A. 1952).
Robert Peter Melson, B.Sc.(Eng.) (A. 1952).
Gordon Douglas Sharpe, B.Sc.(Eng.) (A. 1953).

Student

Neil Cameron Mackintosh (A. 1953).